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CONTRACT TRIAL OF THE UNITED STATES PROTECTED CRUISER *ALBANY*.

BY LIEUTENANTS HAROLD P. NORTON AND ASSISTANT NAVAL CONSTRUCTOR H. G. GILLMOR, U. S. NAVY.

The U. S. S. *Albany* is a steel, sheathed, twin-screw, protected cruiser built by Sir W. G. Armstrong, Whitworth & Co., from designs prepared by them. The vessel was one of three contracted for by the Brazilian government in November, 1894. In March, 1898, prior to the commencement of the late war with Spain, the United States government purchased from the Brazilian government the *Amazonas* (*New Orleans*) then completed, and with their consent acquired by agreement the contract for the *Albany*, then in frame and well advanced.

During the completion of the ship some modifications in her internal arrangements were introduced in order to make her meet more fully the requirements of the U. S. Navy.

Below are given the principal dimensions and data for the ship.

Length on W.L. at trial, feet and inches		345–10
over all, feet and inches		353– 1
Beam at deck, feet and inches		43– 0
water line, feet and inches		43– 6¼
Draft (on trial)	Forward, feet and inches	16– 4⅛
	Aft, feet and inches	17– 4⅛
	Mean, feet and inches	16–10⅛
Displacement on trial, tons		3,450
Deep displacement, tons		4,011
Tons per inch at trial, water line		23.01
Moment to alter trim one inch, ton feet		392.4
Area of trial, water plane, square feet		9,668
Center of gravity of trial plane abaft midship section, feet		5.5
buoyancy above bottom of keel, feet		9.92
shaft midship section, feet		3.00
gravity above keel		16.30
Transverse metacenter above C.B. (B.M.), feet		8.68
Longitudinal metacenter above C.B. (B.M.), feet		453
Transverse metacentric height (G.M.), feet		2.25
Longitudinal metacentric height (G.M.), feet		444.32
Block coefficient of displacement		.505
Midship section coefficient		.887
Coefficient of trial W.L.		.626
Wetted surface to trial W.L.		18,193.14

ARMAMENT.

The main battery of the vessel consists of six 6-inch and four 4.7-inch rapid-fire guns 50 calibers in length. Each gun is mounted on a pedestal mounting and protected by an armored shield moving with the gun. All the guns and mountings of the main battery were manufactured by the builders of the vessel. The secondary battery consists of ten 6-pounder Maxim-Nordenfelt guns, six mounted on cone stands on poop and forecastle and on main deck, and four on port sill mountings under poop and forecastle.

PROTECTION.

The vessel has a complete protective deck, 1¼ inches thick on the flat crown and 3½ inches thick on the sloping sides amidships. The openings for the uptakes, etc., for the boiler rooms are protected by 4-inch glacis plates placed at 45 degrees, worked around the rectangle bounding the whole of the openings above

the boilers. The conning tower is 4 inches thick and has $\frac{3}{4}$-inch flat top and 2-inch shield at entrance. An armored tube, 2 inches thick, extends from the base of the conning tower to the protective deck for the protection of the voice pipes, steering rods, etc.

HULL.

The hull is constructed of mild steel throughout. The frames are Z-bar spaced generally four feet from center to center excepting where for special purposes intermediate frames have been introduced.

The ship is sheathed with one thickness of teak $3\frac{1}{2}$ inches thick, secured by bolts of naval brass tapped into the steel skin of the ship and having naval-brass nuts set up upon their points on steel washers inside the ship. Between the wood sheathing and the steel skin thick paint has been injected under pressure. Upon the outer surface of this wood sheathing there are three coats of coal tar and two layers of heavy tarred paper and over the whole, up to a height of 12 inches from top of wood sheathing, 24-ounce copper is secured with sheathing nails. Below the protective deck there are fifty water-tight compartments, excluding those of the double bottom. The double bottom extends between frames No. 17 and No. 141 (frames being numbered on a basis of two feet spacing), and is separated into twelve compartments. Of these four have been fitted for the stowage of fresh water for reserve feed and any of the others may be filled with sea water for ballast if desired. The accompanying plan will give some idea of the arrangement of the vessel.

MAIN DRAIN.

For dealing with large volumes of water in engine or boiler rooms there is a 12-inch main drain near the middle line, above the inner bottom, extending from the after end of forward boiler room to the after end of engine room, with 12-inch suction branches fitted with 12-inch screw-down non-return valves, operated from the level of the berth deck in each engine and boiler room. It has also 12-inch branches to each circulating pump of main condensers fitted with 12-inch valves. Water

drawn from the main drain is discharged overboard through the main condensers and sea valve No. 18 or sea valve No. 19. There is a 3-inch branch between manifold No. 2 and the main drain in the after boiler room to permit it to be flooded from the sea through manifold No. 2, and by this branch any of the pumps on the secondary drain may be used on the main drain as well.

SECONDARY DRAIN.

For the ordinary flooding or pumping of compartments below the protective deck, there is a secondary drain above the inner bottom near the middle line extending from the after end of the starboard engine room to the forward end of forward boiler room. From forward end of engine room to after end of forward boiler room the drain is 6 inches in diameter, elsewhere 4 inches in diameter. There is a direct suction on secondary drain in each boiler room fitted with a screw-down non-return valve. The steam pumps connect directly to the secondary drain in the engine room and boiler rooms. The hand pumps connect with the secondary drain through the manifolds. The secondary drain is connected to five manifolds by means of which the pumping of the compartments with either the steam or hand pumps is made possible. There are twenty-five sea valves for use in connection with the general pumping and drainage of the vessel, all of which, excepting those in the machinery space, are operated from the berth deck through locked indicating deck plates. Five high-duty steam pumps of the vessel are arranged for use on the general drainage system. In addition to these there are three large hand pumps which may be employed for the pumping of any compartment of the vessel. Through the manifold and the secondary drain suctions from them, any of the compartments, to which suction branches are led from the manifolds, may be pumped by the steam pumps having suction connections to this drain or by the hand pumps. By opening the sea-flood valve and the drain-suction valve at the manifold, the compartment may be filled through the suction branch to it. For the purpose of permitting the drainage of water from compartments not provided with direct drain suctions into those

which have such suctions in order that it may be removed by the pumps, drain valves are fitted in several compartments.

FIRE SYSTEM.

A 4-inch fire main of seamless copper extends fore and aft, under the main deck from frames No. 43 to No. 131, with 4-inch risers at these frames to forecastle and poop. From this main there are branches with 2½-inch hose connections for wash-deck and fire purposes in berth-deck space on upper poop and forecastle decks and under poop and forecastle. Each of the steam pumps, except the main-feed pumps, may supply the fire main. The two auxiliary feed pumps have each a 3-inch riser, discharging into the fire main, and the fire and bilge pumps (three in number) in the engine rooms discharge into a 4-inch riser at about frame 103. There are several 4-inch cut-off valves in the fire main, so that it may be cut into several sections which may be used independently.

Each of the three hand pumps is fitted with a connection to the fire main, and the piping and valves are so arranged that each pump may draw from the sea and discharge into the fire main, overboard, or into a hose attached to the pump or draw from the fire main and discharge either overboard or into a hose attached to the pump. By this means hose attached to any of the fire or wash-deck connections of the fire system may be employed as suctions for any compartment above the protective deck and so pumped out by the hand pumps.

FRESH-WATER SERVICE.

For the stowage of fresh water required for general purposes there are two tank compartments, A5, capacity, 7,375 gallons, and D12, capacity, 4,024 gallons. Four compartments of the double bottom, B94, B95, capacity, 6,777 gallons each, and B96, B97, capacity, 5,877 gallons, are arranged for the stowage of fresh water for reserve feed for the main boilers. Fresh water from the ship's distillers is discharged into the test tank at the side of boiler casing amidships on the starboard side, whence it flows by gravity into the ship's tanks, A5 and D12, or into the

reserve-feed tanks through direct pipes led to these tanks and provided with cocks, properly labeled to indicate their purpose. The ship's tanks, A5 and D12, may also be filled from water boats alongside, there being abreast each tank, forward and aft, a water-boat connection, outboard on each side, fitted with a screwed cap and having inboard, in addition, a cut-off cock. By coupling the 1½-inch hose provided to any of the water-boat connections the tanks may be filled by the water-boat pumps, or the ship's fresh-water pumps may be used to pump water from the water boat into the ship's tanks. The reserve-feed tanks may be filled from water boats through the filling pipe on main deck amidships or by means of a hose connection on this pump it may be employed to pump water into the reserve feed tanks.

For the distribution of fresh water to the several points in the ship at which it is consumed, there is a system of piping. This system is connected with two gravity-distribution tanks, one of which is underneath the chart house at the after end of the forecastle, and one at the after end of the funnel casing over the officers' galley. There are two 3-inch lift pumps, one forward and one aft, which draw from the ship's tanks and discharge into the gravity tanks. By gravity flow the water is distributed through the piping system and its branches to the pantries, galleys, bath rooms, etc., in which it is required.

FLUSHING SYSTEM AND SALT-WATER SERVICE.

The supply of salt water for flushing and general purposes in the ship is distributed through a system of piping. The mains are 2½ inches in diameter from the riser from the fire and bilge pumps in the engine room to the head, elsewhere, 1½ inches. In connection with this piping system there are two gravity-distributing tanks abreast the two tanks mentioned in connection with the fresh-water system. Ordinarily the circulating water for the distilling apparatus will be discharged into the flushing system and so overboard, serving to supply salt water for general purposes and for flushing the heads and water closets. When the distillers are not in use or it is not desirable to use their circulating pumps on the system, by closing the

two cut-off valves on the fire system at the riser from the engine room pumps and opening the valve in the branch connecting this riser with the flushing system any of the three pumps in the engine rooms may be used on the flushing system. When it is undesirable to employ any of the steam pumps mentioned, the gravity-distribution tanks may be filled by two 3-inch salt-water lift pumps, provided for the purpose, and the distribution to the system from these tanks is then by gravity flow. Flushing tanks have been fitted for the Captain's and officers' water closets. The sick bay and warrant officers' water closets, forward, are flushed from the forward gravity tank.

GENERAL VENTILATION.

The ventilation of the spaces below the protective deck is divided up among virtually four systems, two forward and two aft, the two forward being connected with one another, as are also the two aft, though there is no connection between the forward system and the after ones. On the berth deck forward there is a steam fan on the port side and an electric fan on the starboard side on bulkhead No. 41. These fans draw fresh air directly from a ventilator with cowl top on the forecastle and discharge into separate ducts led down and through the protective deck, for the ventilation of the lower compartments on the two sides. Large sluice valves are fitted to the openings in the protective deck where each of these ducts cuts it. Similarly at the after end of the engine room on bulkhead No. 117 there is a steam fan on the starboard side and an electric fan on the port side, drawing fresh air from the after engine-room ventilators and each discharging into ducts, for the ventilation of the compartments on the side on which the fan is. The forward systems are connected across below the protective deck at frame No. 38, and the after systems at about frame No. 120, so that in both sets of systems either fan may be used for the ventilation of either side, or either fan for the ventilation of both starboard and port sides. Where ducts cut bulkheads, automatic ball valves are fitted to be self-closing in case the compartment on either side of the bulkhead should be flooded. For supplying fresh air to

storerooms, etc., 4-inch branches are taken off from the ventilating ducts at an angle of 45 degrees and discharge directly into the compartments.

From the ventilation ducts above 4-inch branches are led to the magazine and shell rooms, terminating at the bulkheads, branches being led to the main magazines from both the starboard and port systems and single branches only to the smaller ones. These branches are led in as low down as practicable, and supply fresh air to the spaces. Four-inch exhaust pipes are led from the crown of the magazines and shell rooms to about the height of the main-deck beams, where they are fitted with goose necks and perforated plates to prevent anything being introduced through them.

Fresh air is supplied to the lower bunkers by 6-inch ventilating pipes, which lead from the level of the main deck, inside the downcasts, to the forced-draft fans, and discharge through openings in the bulkheads of the cross bunkers, one on each side, making one supply pipe for each bunker compartment. Exhaust pipes, 6 inches in diameter, are led from each end of each wing bunker, into which the cross bunkers open, into the funnel casing, discharging at the height of the main-deck beams. The air supplied to the cross bunker, therefore, finds its way to the ends of the wing bunkers, and is withdrawn and discharged through the exhaust pipes into the funnel casing. All the supply and exhaust ducts of the ventilation system of the lower bunkers have sluice valves fitted with hand wheels and indicating gear at the bunker bulkheads. For the ventilation of the bunkers above the protective deck, supply and exhaust ducts are led from the topsides to the crown of these bunkers, so arranged that the air, entering by the supply ducts, will traverse the bunker and leave by the exhaust ducts. All these ducts are fitted with special louvres in the topsides.

The paint and oil store is provided with a special ventilation pipe extending from the flat between A3 and A4 to the forecastle deck, where it is fitted with a mushroom top.

The ventilation of the engine rooms and steering compartment is by natural means, fresh air being supplied from downcast

ventilators, exhaust being by hatch trunks. The fire rooms are supplied with fresh air from downcast ventilators, to each of which is fitted a steam fan for the production of forced draft, exhaust being through the furnaces and funnels and through the entrance shafts. The ventilation of the entire living space and all compartments above the protective deck is by natural means.

STEERING ARRANGEMENTS.

The rudder is balanced and so hung that its weight is taken on the rudder-head and stuffing-box casting. The steering is effected through a tiller, Rapson's slide, auxiliary head and parallel bars. A sprocket chain, shackled to the sliding block on the tiller, is actuated by a sprocket wheel driven by the steam-steering gear or the hand gear. A tightening roller is fitted on each side, provided with a right-and-left-handed screw, so that the tension of the sprocket chain may be properly adjusted. An oil cup on each side is fitted in such a way as to drop oil upon the links of the sprocket chain.

The steering engine was built by Davis & Co. of Garford street, London, and is of a pattern commonly employed in Admiralty vessels. The valve arrangements of this engine are of the usual type for such engines, but there is fitted to the engine an automatic control valve by which the supply of steam to the engine is automatically increased as the rudder angle increases. This valve may be roughly described as a throttle valve actuated by a screw driven by gearing from the engine, so that at small angles of rudder the steam is very much throttled, while as the angle increases the control valve opens, and so increases the supply of steam to the engine in proportion to the additional work required to move the rudder at large angles.

The vessel may be steered by steam-steering wheels from the fore bridge and the conning tower, the motion being transmitted by rods and gearing led through the conning-tower tube to below the protective deck and thence aft. Steering by steam is also possible from the steering-engine room by a steering wheel fitted on the engine. The vessel may be steered by hand by the steering wheels under the break of the poop. The effort is trans-

mitted by shafting and gearing actuating the sprocket wheel mentioned above. To shift from steam to hand steering there is a clutch operated by a hand wheel and screw on the port side at the after end of the steering engine. The change from one means of steering to the other may be effected only in the steam-steering engine room.

COAL BUNKERS.

The general arrangement and location of the coal bunkers is indicated in the annexed diagram. The capacities given in the diagram have been calculated from the dimensions taken from the ship, the stowage being reckoned at forty-three cubic feet per ton and coal assumed to be stowed within six inches of the bottom of the beams in all cases. The calculated capacities are, for the upper bunkers, 388 tons, and for the lower bunkers, 360 tons; total, 748 tons.

All bunkers are coaled from the main deck only, so that the ship must be coaled "over all." Through scuttles in the main deck and trunks to the upper bunkers these bunkers are filled. Access to and egress from these bunkers is also by way of the scuttles and trunks, though it may also be had through the trunks to lower bunkers. The doors between the upper bunker compartments are all hinged doors, and all are fitted with manholes opening the opposite way from the door itself, so that access may always be had from one bunker to that adjacent to it. There is an overhead rail (with portable lengths at each door) and trolley, from which is suspended a bucket or carriage for the transporting of coal from one compartment to another, for trimming in the upper bunkers.

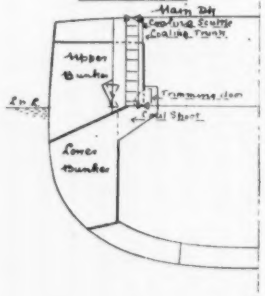
The lower bunkers are coaled through scuttles in the main deck, and trunks lead thence through the protective deck to the bunker. Each lower bunker has at least two such scuttles and trunks, and the middle bunkers, B5 and B6, have three each. The openings in the protective deck at these trunks are all fitted with armored covers. The coaling trunks for the lower bunkers are provided with hinged scuttles or chutes, opening into the upper bunkers, and coal is trimmed to any of the lower bunkers

Capacities of Upper Bunkers

A 115	13 Tons
A 116	13 "
B 103	00 "
B 104	00 "
B 109	46 "
B 110	46 "
C 103	57 "
C 104	87 "
D 107	18 "
D 108	18 "
Total	388

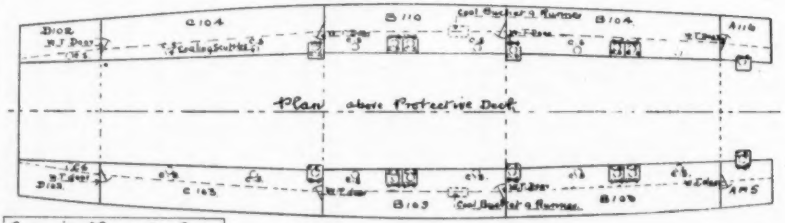
Section showing Trimming Arrangement

Scale 8" = 1 Ft



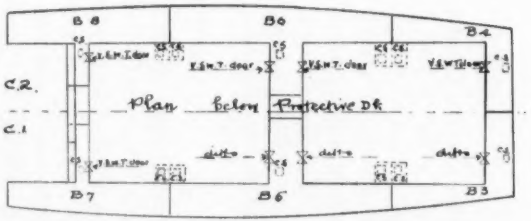
Sketch of Bunkers shewing Coaling & Trimming Arrangements

Scale 16" = 1 Ft



Capacity of Lower Bunkers

B 8	43 Tons
B 4	43 "
B 5	66 "
B 6	84 "
B 7	53 "
B 8	50 "
Lower Bunkers Total	359 "
Upper Bunkers	388 "
Total Capacity	747 "



COAL BUNKERS.

through these chutes and the trunks, there being the same number of openings for trimming as for coaling the bunkers.

CAPSTAN AND WINDLASS.

The ship is provided with a combined capstan and windlass driven by a vertical, twin-cylinder, simple steam engine. The windlass is horizontal and so arranged that the two cables may be worked independently of one another. The capstan on the forecastle is driven by the same engine, a suitable arrangement of clutches and gearing making it possible to use it independently of, or in connection with, the windlass at slow or fast speeds. Whelps on the capstan are fitted to take the cat chain.

ELECTRICAL PLANT.

The vessel is provided with three 32-Kilowatt dynamos manufactured by the builders of the vessel. They are of the four-pole type having cast-steel field magnets with detachable pole pieces. The machines are compound wound and arranged so that the several generators may work in parallel. These dynamos are driven by two-cylinder, compound, vertical engines, 8¼ inches by 14 inches by 8 inches stroke, on the same bedplate with the dynamo. The dynamos and engines are designed to work at 325 revolutions per minute. Two of these dynamos are placed in one compartment adjacent to and opening into the engine room on the starboard side aft, and the third in a similar compartment on the port side. The switchboard is placed in the same compartment with the one dynamo on the port side. The dynamo rooms are connected by a voice pipe. Access to these dynamo rooms may be had from the main deck or from the engine rooms. Each dynamo is connected to the main switchboard. The ship is wired on the two-wire system, steel conduit being used throughout the watertight system, and wood casing, in the living spaces and quarters, for protecting the wiring. There are three searchlights, constructed by the builders and fitted with metallic mirrors manufactured by the Cowper-Cowles process. These mirrors are of copper upon which is electrically deposited silver, upon which is again deposited, to protect from

oxidation, etc., a thin film of palladium. The searchlights are fitted with combined hand and automatic feed.

MAIN ENGINES.

There are two three-cylinder, triple-expansion engines of the vertical, inverted, direct-acting type, placed abreast of each other, in separate watertight compartments, the H.P. cylinders being forward.

The main engines and boilers were designed and built by R. & W. Hawthorn, Leslie & Co., Ltd., St. Peter's Works, Newcastle-on-Tyne, although a number of changes in the original design have been made to make it conform to U. S. naval practice.

The engine framing consists of cast-steel inverted Y-frames at the front, and forged-steel columns at the back or outboard side.

The cylinders are tied together by a forged-steel tie rod on each side, and are braced athwartships by a stay from each cylinder to the center line and outboard bulkheads of the engine rooms.

There are two ramming stays, consisting of short wire pennants with adjusting screws, fitted between each L.P. steam chest and the protective deck.

The H.P. valve is of the piston type and those for the I.P. and L.P. are double-ported slides. All the valves are fitted with balance pistons.

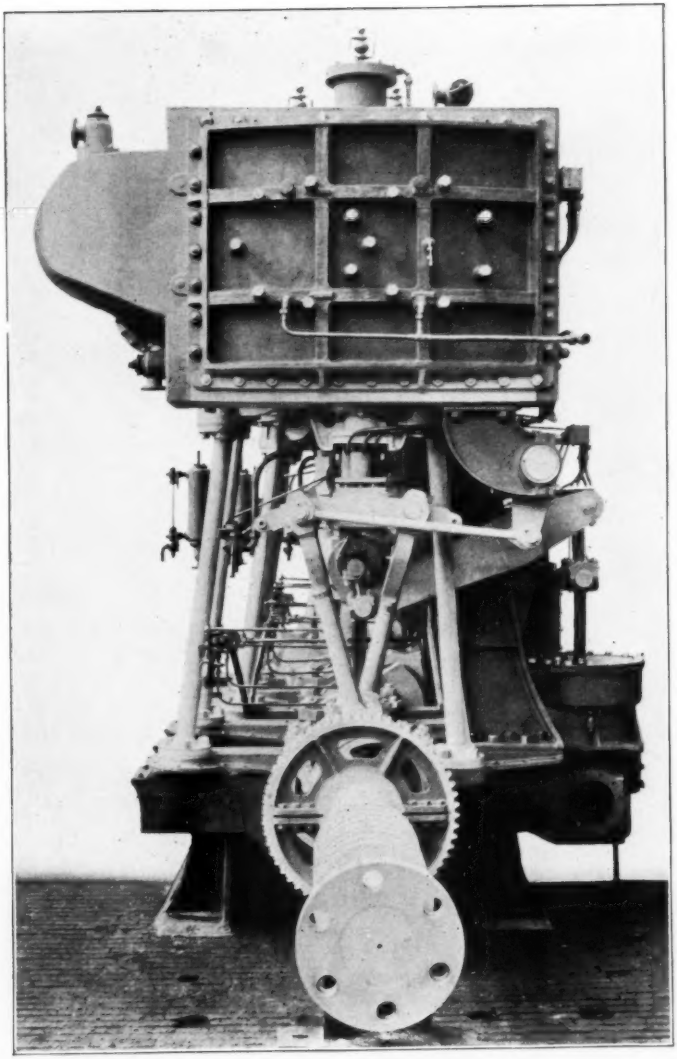
The link motions are of the double-bar, Stephenson type, and the valve gears for the three cylinders are interchangeable.

Independent cut-off blocks are not fitted in the reversing arms for the links, and to change the cut-off in one cylinder it is necessary to change it in the other two.

There is a starting valve on the I.P. cylinder, which admits live steam to the steam passages of that cylinder and into the exhaust pipe leading to the L.P. steam chest.

The I.P. and L.P. slide valves are fitted with balance rings on their backs to relieve the pressure on the valve faces and seats.

PORT MAIN ENGINE, ERECTED IN SHOP.



PORT ENGINE—END VIEW.

All the cylinders are steam jacketed around the working linings.

Each engine has a combined stop and throttle valve, of the balance double-disc or double-poppet type. This is operated by a direct screw with the hand wheel at the starting platform.

For handling the engines quickly and running slowly there is a small disc valve fitted on the throttle-valve casing, which allows the steam to pass around the throttle valve when the latter is closed. This by-pass or manoeuvering valve is worked by a direct screw with the hand wheel at the starting platform.

The main pistons are steel castings, dished in form. The H.P. and I.P. are fitted with three spring packing rings, the L.P. being fitted with one broad packing ring and springs for setting it out. All the pistons are fitted with followers.

The piston and connecting rods are of steel. The crossheads are fitted with composition shoes, with white metal bearing surface for the ahead guides. The bearing surface when backing is formed by two composition plates fitted on the back of the crossheads and bearing on the backing guides, which are made of cast iron and bolted to the back columns.

The crosshead guides are of cast iron, and are secured to the invered Y-frames, the ahead guides being fitted for water circulation to keep them cool.

The eccentrics are of cast iron and cast steel, the smaller portion being of cast steel. Each eccentric is secured to the crank shaft by a steel key and one set screw.

The eccentric straps are of composition faced with white metal. The eccentric rods are of steel, secured to the straps by a T-head and bolts.

The valve stems are guided by the stuffing boxes, which are made deep and are well braced by ribs.

The engine bed-plates consist of six steel castings which are secured by bolts to the six steel seatings, which are built into the hull works by the shipbuilders. Each bed-plate casting has a seating for one of the forged-steel back columns, one foot of the inverted Y-frames at the front and at its center is so designed

and fitted as to form the lower portion of one of the crank-shaft bearings.

The reversing gear is of the "all-around" type, and consists of a double-cylinder, simple engine, with the usual worms and worm wheels, hand wheel and regulating valve. The regulating valve and hand wheel are conveniently placed at the starting platform, the hand wheel being fitted with a key so that it may be disconnected, if desired, when the steam gear is in use.

The piston rods and valve stems are fitted with metallic packing. This packing consists of a series of white metal and composition rings.

The first ring that is placed in the box is fitted with a series of spiral springs fitted in recesses in the ring and pressing against the bottom of the box. Between the gland and the last ring of metallic packing several turns of square Tuck's packing are placed. In addition to the above the glands for the piston rods are fitted for several turns of Tuck's square $\frac{3}{8}$-inch packing.

The nuts of the stuffing-box glands are fitted with gear for setting them all up at the same time.

There is a turning engine with one cylinder $6\frac{1}{4}$ inches diameter by 6 inches stroke, secured to the after section of the bedplate of each engine. This works into the worm wheel on the after end of the crank shaft.

A hand lever is also fitted so that the engines may be turned by hand when there is no steam, or for making adjustments of the valve gear.

The main steam pipes, 12 inches diameter, are of steel, lap welded and butt strapped and riveted, and are fitted with forged-steel flanges riveted to the pipes. The bends, slip joints and junction pieces are of composition. The auxiliary steam pipes are of solid-drawn copper with brazed flanges, and the auxiliary exhaust pipes are of brazed copper with brazed flanges.

The main steam pipes are separate for each pair of boilers. The branch pipes from the two boiler stop valves in the same compartment are connected by junction pieces and slip joints to the 12-inch steel main steam pipe for these two boilers. This pipe is carried direct to one of the engine rooms. The two forward

boilers are connected to the port engine and the two after boilers to the starboard engine, with a stop valve in each at the forward engine-room bulkheads. The main steam pipes in the two engine rooms are connected by an athwartship pipe with a valve on the fore and aft bulkheads, which valve can be operated from either engine-room. This allows both engines to be run from the boilers in either compartment, without the necessity of having steam in the main steam pipe for the boilers not in use.

There is an auxiliary steam pipe extending from the steering engine at the stern to the capstan at the bow.

This auxiliary steam pipe has a connection to each boiler and a connection in each engine room to the main steam pipe. The auxiliary steam and exhaust pipes are fitted with valves wherever they pass above the protective deck so that in action all steam, except for whistle and siren, may be shut off from above that deck.

There is a large separator in each main steam pipe, placed close to the throttle valve; this has a drain and trap with by pass discharging into the main feed tank.

All the steam and exhaust pipes are fitted with drain pipes. There is a two-inch pipe placed below the floor plates and extending the entire length of the engine and boiler compartments, with a discharge into the port condenser, reserve feed tanks and the bilge. The several drain pipes from the steam pipes are connected and deliver into traps which discharge into the above-mentioned two-inch pipe. The several drain pipes from the exhaust pipes are similarly connected, and discharge into the same two-inch pipe.

To prevent water backing up into any of the steam or exhaust pipes through the drain pipes there is a check valve at each connection, and at each drain valve on the steam and exhaust pipes there is a test cock, so that it is always possible to determine if the drains are working properly. This arrangement allows the drain pipes to be always open when a pipe is in use and ensures its being free of water.

The auxiliary exhaust pipe connects with the main condensers, the auxiliary condenser and the atmosphere, the atmos-

pheric valve being automatic, so that should it be necessary to shut off the exhaust to the condenser, or should the pressure rise in the exhaust pipe due to the failure of the pumps, the exhaust to the atmosphere will open by the increased pressure.

There are two twenty-gallon oil tanks for supplying the oil, under a pressure, for filling the oil boxes on the engines. These tanks are placed in the engine hatch, and each may be filled from the oil tanks in either engine room, a small rotary pump being placed in each engine room for this purpose.

The oil boxes on the engines are sight-feed wick cups, the amount of oil used depending on the size of the wick. Each tube for the wick is fitted with a cock so that the supply of oil may be shut off without removing the wick. The crank pins are fitted with centrifugal oiling gear and with the usual oil box and tube.

SHAFTING AND BEARINGS.

All the shafts are of forged steel and, with the exception of the propeller shafts, are solid.

The crank shafts are made in three sections, those for the I.P. and L.P. being interchangeable. The cranks are 120 degrees apart, and for ahead motion the sequence is, H.P., I.P. and L.P. There is a 5¾-inch axial hole through the propeller shafts. These shafts are each in one section, 66 feet 3 inches long, the outside diameter being variable. They are fitted with composition casing from inboard of the stern-tube stuffing box to the propeller hub, where the casing makes a watertight joint. The axial hole in the propeller shafting is tapered where it passes through the propeller hub.

The casing is fitted in long sections which do not have a bearing on the shaft for their entire length. The ends of the sections of the casing are halved where they lap, and a dovetailed groove is turned partly into each of the adjoining sections. This groove is run full of tin, which when cool is rolled by a tool so that it completly fills the groove. The spaces between the shaft and casing are then filled with a solution of red lead and oil, forced in by a pressure pump.

All the shafts run in white-metal bearings. The thrust bear-

ing is of the horse-shoe type, the collars, eight in number, running in an oil bath. There is a bearing at each end of the thrust bearing to take the weight of the shaft.

The horse shoes are of cast steel and are faced with white metal, each shoe being adjusted independently of all the others by means of the screw and nuts on each side of the pedestal. The bearing works in a bath of oil and water.

ENGINE DATA.

Cylinders, number for each engine	3
H.P., diameter, inches	31
I.P., diameter, inches	46
L.P., diameter, inches	70
Stroke of pistons, inches	30
Valves, H.P. piston (one for each cylinder), diameter, inches	18
I.P. slide (one for each cylinder), diameter, feet and inches,	3- 6⅛ × 3- 7½
L.P. slide (one for each cylinder), diameter, inches	43⅛ × 66½
Balance pistons, H.P., diameter, inches	5
I.P., diameter, inches	6
L.P. diameter, inches	8½
Valve stems, H.P. (1), diameter, inches	4
diameter through valve, inches	2¼
I.P. (1), diameter, inches	4
diameter through valve, inches	2¼
L.P. (1), diameter, inches	4
diameter through valve, inches	2¼
Main steam pipe (12 inches diameter), area cross section, square inches	113.1
Exhaust pipes to I.P. cylinder (1), 16 inches H.P. end, 16 inches I.P. end, area of cross section, square inches	201.1
Exhaust pipes to L.P. cylinder (1), 20½ inches I.P. end, 20½ inches L.P. end, area of cross section, square inches	330
Exhaust pipes to condenser (1), 23½ inches diameter, area of cross section, square inches	433.7
Volume swept by H.P. piston (mean) per stroke, cubic feet	12.88
I.P. piston (mean) per stroke, cubic feet	28.63
L.P. piston (mean) per stroke, cubic feet	66.58
Net area of I.P. to H.P. piston	2.223
L.P. to I.P. piston	2.326
L.P. to H.P. piston	5.170
Clearances of H.P. cylinders, per cent	top, 18; bottom, 18
I.P. cylinders, per cent	top, 16; bottom, 16
L.P. cylinders, per cent	top, 15; bottom, 15
Piston rods, diameter, inches	5¼
length from piston to crosshead, feet and inches	4- 3

Connecting rods, length from center to center, feet	5
diameter of upper end, inches	5½
lower end, inches	6
crosshead bolts (2), diameter, inches	3¼
crank-pin bolts (2), diameter, inches	3¼
Crossheads, surface (ahead), for each crosshead, square inches	387.5
(backing), for each crosshead, square inches	240.0
pins, diameter, inches	7
length, inches	14
Reversing gear, steam cylinders (2), diameter, inches	6
stroke, inches	5
Crank shafts, solid, forged steel, diameter, inches	11½
coupling discs, diameter, inches	22½
thickness, inches	2¾
bolts in one flange (6), diameter, inches	2⅝
journals, diameter, inches	11½
length, inches	15
length of each section, L.P. and I.P., feet and inches	5– 9⅝
H.P., feet and inches	7– 2½
Crank pins, diameter, inches	12
length, inches	14
Crank webs, width, inches	14
thickness, inches	8
Thrust shaft, solid, diameter, inches	11⅛ and 11½
of coupling disc, inches	22½
collars, number each shaft	8
diameter, inches	18
thickness, inches	1½
distance between, inches	2⅞
surface, total for both engines, square inches	1,792
length, feet and inches	10– 0
Line shafts, solid, diameter, inches	11
(2), length, each, feet and inches	14– 2½
diameter of coupling discs, inches	22½
Propeller shafts, propeller end, diameter, inches	11½
center, inches	13 15/32
forward end, inches	11½
axial hole, inches	5¾
Propeller shafts, outboard, length, feet and inches	66– 3
coupling diameter, inches	22½
Stern bracket bearing, length, inches	60
diameter, inches	13¾
tube bearing, length, forward, feet and inches	3– 9
diameter, forward, inches	13⅝
length, aft, feet and inches	4– 5
diameter, aft, inches	13½

MAIN CONDENSERS.

There are two main condensers, one for each engine. They are made of composition, all parts being cast, the shell being made in sections with flanges for bolting it together. The shells are made with flat sides with hemispherical top and bottom.

The circulating water enters at the top of the condenser, passes through the tubes and out at the bottom. The tubes are $\frac{5}{8}$-inch diameter and No. 18, L.S.G. Diaphragms are fitted in the condenser heads so that the water passes through the condenser twice before it reaches the outlet. There is a fresh-water "make-up" from the reserve-feed tanks and a salt-water "make-up" on each condenser.

CONDENSER DATA.

Diameter of shell (inside), feet and inches	8-8 × 3-9
Thickness of shell, inch	$\frac{7}{16}$
Length over heads, feet and inches	9-1$\frac{1}{4}$
Tubes, diameter (outside), inch	$\frac{5}{8}$
length between tube sheets, feet and inches	7-6
thickness, No. 18 L.S.G., inch	.048
number in each condenser	3,674
Cooling surface, each condenser, square feet	4,500
Total cooling surface, square feet	9,000
Ratio of total cooling to total heating surface	.684

MAIN AIR PUMPS.

There is one vertical, single-acting air pump for each engine. These pumps are placed on the inboard side of the engine and are worked off the L.P. cross-head by levers.

The pump valves are made of soft rubber, being segments of a circle 20$\frac{5}{8}$ inches diameter at the outside and 6$\frac{5}{8}$ inches diameter at the inside.

MAIN AIR-PUMP DATA.

Diameter of pump cylinders (1), inches	22
Stroke, inches	14
Diameter of pump rods, inches	2$\frac{3}{4}$
Ratio of volume swept by L.P. piston, net area per stroke to that of the two air pump buckets per stroke	10.9

MAIN CIRCULATING PUMPS.

There is one double-inlet, centrifugal circulating pump for each main condenser, which is arranged to draw from the sea and the main drain pipe and discharge into the main condenser.

The main injection valve is on the ship's side, and the outboard delivery is on the ship's floor or bottom, directly under the condenser, the circulating water entering the condenser at the top and discharging at the bottom.

The main condensers are abreast of each other, with the fore-and-aft water-tight engine-room bulkhead between them, and connecting pipes and valves are fitted so that either pump can supply either or both condensers.

The circulating pumps and engines were manufactured by W. H. Allen & Co., and the pumps have each a capacity of 1,328 gallons per minute.

MAIN CIRCULATING-PUMP DATA.

Diameter of steam cylinders, inches	7½
Stroke, inches	5½
Diameter of pump runner, inches	30
Width of runner, inches	6½
Diameter of inlet and of outlet nozzle, inches	12
Capacity per minute	both pumps 10 tons or 2,656 gallons.

AUXILIARY CONDENSERS.

There is one auxiliary condenser, placed in the starboard engine room. It is made of composition, with flat sides and hemispherical top and bottom. The tubes are ¾ inch outside diameter and No. 18 B.W.G. in thickness. It is fitted with a double-inlet, centrifugal, circulating pump and a vertical, single-acting air pump, both of which are placed directly under the condenser, and are worked by a vertical, inverted, single-cylinder engine.

The circulating pump is on the engine shaft, and the air pump is worked from a pier in a disc at the end of the shaft.

The auxiliary condenser, engine and pumps were manufactured by Earnest Scott and Mountain.

AUXILIARY-CONDENSER DATA.

Cooling surface, one condenser, square feet	300
Diameter of steam cylinders, inches	5
circulating-pump runner, inches	15
air-pump cylinders, inches	6
Stroke, steam cylinder, inches	4
air pump, inches	4

SCREW PROPELLERS.[1]

The propellers are right-and-left handed, of the modified Griffith type, of composition, each with three adjustable blades, the surfaces of which are polished. They are true screws, with pitch adjustable from 14 feet 9 inches to 16 feet 9 inches.

Each hub is secured to the shaft by one feather key and a nut screwed on and locked in place. The nuts are covered by a conical composition cap, and the recesses for the bolts securing the blades to the hub are fitted with composition plates.

The joint between the hub and the composition casing on the propeller shaft is made tight by a rubber gasket set up by a composition ring and bolts. The casing is recessed into the hub.

The propellers are so placed on the shaft that each blade bisects the angles between the cranks of the engine.

SCREW-PROPELLER DATA.

Number of blades	3
Diameter, feet and inches	12-0
of hub, feet and inches	3-3
Length of hub, feet and inches	2-5
Pitch as set, feet and inches	15-8⅛
Greatest width of blade, feet and inches	3-9
Helicoidal area of each screw, square feet	40
Projected area, square feet	31
Disc area, square feet	104.8
Pitch ÷ diameter	1.31

BOILERS.

There are four double-ended steel boilers of the horizontal, return, fire-tube type, with a separate combustion chamber for each furnace.

Each boiler has six Purvis furnaces.

The shell of each is made of three courses of two plates each, the longitudinal joints are treble riveted with double butt straps, the circumferential joints are lapped and double riveted and the heads flanged and double riveted.

The joints in the furnaces and combustion chambers are single riveted.

The furnaces are fitted with wrought-iron grate bars, each bar being separate, and the furnace doors are of the Ashcroft type.

Each boiler has a triple, 4½-inch, spring-safety valve, the three valves being fitted in one case.

The boiler tubes are of lap-welded iron. Each boiler has two hydrokineters, and the auxiliary feed pumps have a connection for pumping out the boiler or for circulating the water when raising steam.

The boilers are placed fore-and-aft in two water-tight compartments, two boilers in each compartment, with an athwart-ship fire room at each end, making a total of four fire rooms.

There is a passageway with air lock from the port engine room to the after fire room, and a passageway between the boilers in the boiler compartments, with an air lock between the two compartments. The working pressure is 155 pounds. There are two circular smoke pipes.

There is a steam whistle and steam siren forward of the forward smoke pipe.

DOUBLE-ENDED BOILER DATA.

Length, feet and inches	18–1⅛
Diameter, outside of all, feet and inches	12–4⅞
Thickness of shell and heads at top, inch	1⅛
heads, inch	1⅛
tube sheets, inch	1⅛
furnaces, inch	1⅛
Combustion chambers, number in each boiler	6
thickness of plates, inch	9/16
depth at top, feet and inches	2–3¾
Furnaces, greatest internal diameter, feet and inches	3–1½
least internal diameter, feet and inches	3–0
length of grate, feet and inches	6–6
number in each boiler (Purvis)	6
Tubes, outside diameter, inches	2½

Tubes, length between tube sheets, feet and inches	6–3⅝
number of ordinary, each boiler	512
stay, each boiler	132
spaced vertically, inches	3½
horizontally, inches	3½
thickness of ordinary, L.S.G., No. 9, inches	.144
stay, inch	½
Diameter of rivets in shell sheets, inches	1½
screw stays, inches	1⅝
Number of through upper braces, 10; diameter, inches	2½
braces from head to back tube sheets, 4; diameter, inches	1½
Heating surface, tube, square feet	2,660
plate, square feet	629
total, square feet	3,289
Grate surface, square feet	117
Area through tubes, square feet	16.6
over bridge walls, square feet	20.5
Volume of furnaces and combustion chambers above grates, one boiler, cubic feet	447
Steam room, 4 inches above highest heating surface, cubic feet	461.7
Water surface, 4 inches above highest heating surface, cubic feet	181.6
Smoke pipes (2), diameter, feet and inches	7–0
area of both, square feet	76.96
height above lowest grates, feet and inches	60–0
Number of safety valves, 3, on one base; diameter, inches	4½
Diameter of boiler main-stop valve (one), inches	8½
auxiliary stop valve (one), inches	3½

TOTAL, ALL BOILERS, DATA.

Heating surface, tube, square feet	10,640
plate, square feet	2,516
total, square feet	13,156
Grate surface, square feet	468
Area through tubes, square feet	66.4
over bridge walls, square feet	82.0
Volume of furnaces and combustion chamber above grates, cubic feet	1,788
steam room, cubic feet	1,846.8
Area of water surface, square feet	726.4
Ratios:	
Tube H.S. to G.S.	22.73
Plate H.S. to G.S.	5.375
Total H.S. to G.S.	28.11
Area through tubes to G.S.	.142
Volume of furnaces and combustion chambers above grates to G.S.	3.81
Steam room per square foot of G.S.	3.946

FORCED DRAFT.

The closed fire-room system of forced draft is used, the entire boiler compartment being under pressure, except the upper portion of the space between the boilers, the air locks for the entrances to the fire room from deck being at the spar deck. The air is supplied by eight fans, manufactured by the builders of the machinery, each fan being driven by a single, vertical, inverted-cylinder, enclosed engine.

Diameter of steam cylinders, inches	7
Stroke, inches	4½
Diameter of fan, inches	60
Width of fan at rim, inches	9

FEED PUMPS.

There are two main and two auxiliary feed pumps, all of the Worthington duplex type as manufactured by the builders of the machinery, with cylinders of 7½ inches by 5 inches by 9 inches, each with a capacity of 150 gallons per minute.

The feed pumps are located in Nos. 2 and 3 fire rooms, one main and one auxiliary pump in each, and only deliver to the boilers in the same compartment.

The auxiliary feed pumps have suctions from the main and reserve feed tanks, sea, secondary drain pipe, the bilge and boilers in the same compartment, with discharges to the boilers in the same compartment, fire main, hose valve for washing boilers, and overboard.

The main feed pumps have suctions from the main and reserve feed tanks, with a discharge to the boilers in the same compartment.

FIRE AND BILGE PUMP.

In each engine room there is a vertical, duplex pump of the same type as the feed pumps, except that it has rubber valves. Cylinders 7½ inches by 5 inches by 9 inches, with a capacity of 150 gallons per minute.

These pumps have suctions from the sea, the secondary drain pipe and the condensers, with discharges to the main feed tank, overboard and the fire main.

In each engine room there is a "Handy Billy" pump, with a suction from the secondary drain, 1½ inches diameter, and a discharge overboard, 1½ inches diameter.

In the port engine room there is a Worthington duplex pump of the Admiralty pattern, with cylinders 6 inches by 4 inches by 6 inches, with a capacity of 60 gallons per minute. This pump has a direct connection to the crank pits in both engine rooms, with a connection to the secondary drain pipe and the sea, with discharges overboard and to the fire main.

GREASE EXTRACTORS.

There is a Rankine's grease extractor fitted in the discharge pipe from each of the main feed pumps. By-pass valves and pipes are fitted, so that the cylinders with the burlap cases can be changed without stopping the feed pump.

FEED TANK.

There is one feed tank of 1,050 gallons capacity, between the engine room and the after fire room. It is built into the hull, and is the same length fore-and-aft as the athwartship coal bunker.

This tank receives the discharge from the main and auxiliary air pumps, the heater, galley and separator traps, and has suction pipes to the main and auxiliary feed pipes. It is fitted with glass water gage, manhole, air pipe to the atmosphere and air overflow pipe to the reserve-feed tanks in the double bottom.

ASH HOISTS.

There is one of Alley and McLellan's ash hoists in each fire room.

The ash hoists are placed in the fire rooms, the ash whips being fitted with stops to prevent over winding.

The engines are not reversible, the empty bucket being lowered by means of a band brake on the drum shaft. Each hoist is worked by two vertical, simple engines with cylinders 4 inches diameter and 5 inches stroke.

STEAM TRAPS.

Steam traps are fitted in the drain pipes from the steam pipes, main separators, heater pipes and steam pipes to the galleys. All traps are fitted with by-pass valves and pipes for use when overhauling.

WORKSHOP FOR THE ENGINEER DEPARTMENT.

The workshop is on the berth deck, inside of the bulkheads for the engine-room hatch, so that it may be entered from the engine rooms. It is fitted with a single-cylinder, vertical, inverted engine, 5 inches diameter by 8 inches stroke, a screw-cutting gap lathe, a shaping machine, a drilling machine, a combined punch and shears, a grindstone and vice benches.

DISTILLING APPARATUS.

The distilling plant is of Kirkaldy's compound compactum type of fifty tons capacity per twenty-four hours. It consists of two evaporators, which may be worked simple or compound, and two condensers.

There are two sets of pumps of the fly-wheel type, with one inverted, vertical steam cylinder for each set, 7 inches diameter and 6 inches stroke. Each set of pumps have the following water cylinders: one for circulating water, $5\frac{1}{2}$ inches diameter and 6 inches stroke; one for fresh water, $2\frac{1}{2}$ inches diameter and 2 inches stroke; one for feed, $3\frac{1}{4}$ inches diameter and 2 inches stroke, and one for brine, $2\frac{1}{2}$ inches diameter and $1\frac{1}{2}$ inches stroke. The circulating water, after passing through the condensers, can be discharged through the flushing pipes or directly overboard.

ICE MACHINE.

There is an Allen dense-air ice machine capable of producing the cooling effect of one ton of ice per day. There is one steam cylinder, 9 inches diameter, one air-compressing cylinder, $5\frac{3}{4}$ inches diameter, and one air-expanding cylinder, $4\frac{3}{4}$ inches diameter, the stroke of all being 10 inches.

The circulating water and primer pumps are single acting, each $1\frac{5}{8}$ inches diameter by 10 inches stroke. The cooling pipes connect to the cold-storage rooms, ice-tank and scuttle butt.

STEAM CUTTER.

There is one single-screw steam cutter, fitted with air tanks. The hull is thirty feet long, and is fitted with one cylindrical vertical boiler of 5.9 square feet grate surface and 68 square feet heating surface. The engine is compound, vertical, inverted cylinder, with cylinders of 4¾ and 9 inches diameter, and a stroke of 5 inches. The air pump and feed pump are worked off the crossheads by brackets so that the stroke of the pumps is the same as that of the main engine.

There is a keel condenser extending on both sides of the keel.

TELEGRAPHS AND REVOLUTION INDICATORS.

There is a Chadburn mechanical telegraph in each engine room connected to transmitters on the bridge and in the conning tower, and mechanical gong in each engine room with a pull at the break of the poop and on the bridge.

Mechanical revolution indicators are fitted in each engine room to show on the same dial the relative speed and the direction of the revolutions of each engine.

In each engine room there is also one of Metcalf's speed indicators of the vertical type, to show at any instant the number of revolutions that the engine is making.

SPEAKING TUBES.

The customary speaking tubes with call bells are fitted between the several parts of the vessel.

OFFICIAL SPEED TRIALS.

The contract of the vessel provided for two speed trials, one under forced draft, during which there should be made four consecutive runs with and against the tide in a calm sea on the mile of 6,080 feet, measured by the British Admiralty, near the mouth of the River Tyne, and a second, to consist of a run of six hours' duration with open fire rooms. The speed under forced draft was specified to be not less than twenty knots per hour, and for the second trial with open fire rooms, a mean speed of at least nineteen knots per hour was required to be maintained.

The forced-draft trial took place on January 9th, 1900, over the measured mile specified. In addition to the complete hull, machinery fittings, boats, rigging, etc., the vessel had her upper bunkers full (about 390 tons), and in the lower bunkers there were about 120 tons of coal. The ship's fresh-water tanks and reserved-feed tanks were completely filled, and in addition considerable ballast of pig iron was on board to bring the vessel to her trial displacement. The weather conditions were favorable, with a fresh W.N.W. breeze blowing with a light surface sea, and no swell. High tide was at about 9.30 a. m. As the vessel had not been subjected to any preliminary trials, some time was taken up in working her up to the full power, and advantage was taken of this to make a number of runs for the purpose of obtaining data for a revolution speed curve for use in connection with the six hours' trial. At 11.40 A. M. full-speed trial was begun, and the series of four runs required by the contract was made without touching the throttle or stop valves of the main engine. The runs were made in pairs, and the mean of the speeds for the four runs was 20.51 knots.

The following is the record of these preliminary and full-speed runs under natural and forced draft.

NATURAL-DRAFT TRIAL RUNS.

Number of run.	Time of starting.	Time on course.	Speed, knots.	Revolutions.	
				Starboard.	Port.
		min. sec.			
Preliminary runs: 1.....	11·0	4 6	14.63	.....	.....
2.....	11·20	3 7	19.25	141½	146½
Official runs:					
1st N.....	11·45	3 13½	18.6	143½	148
2d S.....	12·05	3 2	19.78	148½	151½
3d N.....	12·22	3 2⅝	19.7	150½	154½
4th S.....	12·42	3 1	19.89	149½	154
Mean of means of four runs.....			19.62	150 mean revs.	

FORCED-DRAFT TRIAL RUNS.

5th N.....	1·40	2 56	20.46	154½	159½
6th S.....	1·58	2 58¾	20.20	155½	161½
7th N.....	2·15	2 52½	20.83	158½	165
8th S.....	2·30	2 55	20.57	159½	166
Mean of means of four runs.....			20.51	160 mean revs.	

The throttle valve was not touched from the commencement to the end of the four full-power runs.

On the 11th of January, 1900, the vessel went out for the six hours' trial, being then in practically the same condition as to loading and displacement as upon the first trial.

Advantage was taken of the preliminary runs on January 9th to determine the number of revolutions of the engine to propel the vessel at a speed of 19 knots, so that the six-hour trial could be made without regard to the measured mile.

The weather was very hazy, and after getting clear of the harbor it was found to be very thick and that there was some swell running. Early in the forenoon the fog began to clear to seaward, but, as there seemed little prospect of its becoming sufficiently clear to permit bearings to be taken, it was decided to run seaward, head to sea, and return, maintaining the revolutions at or above 144 per minute, this number of revolutions corresponding to a speed slightly in excess of 19 knots per hour. The trial was accordingly begun at 10 A. M., the vessel running for three hours head to the heavy coast swell and returning during a similar period. The fog several times closed in and threatened to make it necessary to abandon the trial, but the six hours' run was completed without its being necessary to slow down at any time. The mean revolutions during the trial were for the starboard engine 148.2 per minute, and for the port engine 150.6, which, from the revolution speed curve constructed from the results of measured-mile runs, should correspond to a speed of about 19.5 knots per hour. A "Cherub" taffrail log in use during the trial showed a little in excess of 19¼ knots.

During both the trials and during all the time that the machinery was in operation, its performance was very satisfactory in every respect.

Previous to the trial on January 9th the main engines had not been run except for a few hours at the dock and when proceeding from the shipyard to the dry dock.

All bearings were cool, and at no time was water used on any of them. There was no accident to any portion of the machinery. The working of the steering gear and other auxil-

iaries in use during the trials were equally satisfactory. The coal used was the best hand-picked Welsh, but no record of the amount used on the trials was kept.

In determining the number of revolutions necessary for a speed of 19 knots, the engine counters and electrical counter were used. The electrical counter is very similiar to the one designed by Passed Assistant Engineer W. D. Weaver, U. S. N., and used by the Bureau of Steam Engineering in standardizing the screws of vessels.

In the electrical instrument used on the U. S. S. *Albany* the contact for each revolution was made by the valve and gear of one cylinder in each engine room, and the record was made on the paper ribbon by the recording instrument, which was placed in the Paymaster's office on the berth deck The electrical recording instrument was only in use during the time that the vessel was actually on the measured mile.

The I.H.P. of the forced-draft blowers, feed and bilge pumps, circulating pumps, steering engine and dynamos are not included in the following, as they could not be indicated.

CONTRACT-TRIAL DATA, FULL POWER, JANUARY 9, 1900.

Draught of water, beginning of trial, forward, feet and inches	16– 4½
aft, feet and inches	17– 4½
average on trial, feet and inches	16–10⅝
Displacement (mean), tons	3,450
Area immersed midship section, square feet	645
Wetted surface, square feet	18,193.14
Mean speed, knots per hour	20.52
I.H.P., main engines and air pumps	7,083
I.H.P., main engines, air, circulating and feed pumps. (B.) All machinery in use:	
Speed^3 × area immersed midship section ÷ I.H.P. (B)	1,574
Speed^3 × displacement$^{\frac{2}{3}}$ ÷ I.H.P. (B)	547.8
I.H.P. per 100 square feet wetted surface. (B)	38.38
Screw propellers, number	2
number of blades each	3
diameter, feet and inches	12– 0
helicoidal area, square feet	40
pitch on trial, feet and inches	16– 8⅝

SYNOPSIS OF STEAM LOG.

		Starboard.	*Port.*
Slip of screws, per cent		17.93	19.95
Revolutions of main engines per minute		158.8	164.0
	mean	161.4	
Piston speed, feet per minute		747	
Steam pressure (mean) at engines		140.5	141.0
	first receiver, absolute	68.0	69.0
	second receiver, absolute	25.5	25.6
Vacuum in condenser, inches		27.0	24.7
Throttle-valve opening		5.75	4.9
Steam cut-off in fraction of stroke from beginning,	H.P	21.75	21.9
	I.P	21.75	21.9
	L.P.	22.62	22.64
Temperatures in degrees Fahrenheit,	Engine rooms	73	70
	Injection	47	46
	Discharge	102	108
	Feed at tank	99	99
Air pressure in fire rooms in inches of water		1.35	
Mean effective pressures,	H.P	53.96	56.60
	I.P	33.23	32.26
	L.P	13.05	12.14
Aggregate equivalent pressure referred to L.P. piston		37.78	36.96
Revolutions or double strokes per minute,	Main air pumps	158.8	164
	Main circulating pumps	208	244
	Main feed pumps, average	61	
	Eight forced-draft blowers, av'ge	382	
I.H.P.,	H.P. cylinder	978	1,048
	I.P. cylinder	1,338	1,329
	L.P. cylinder	1,222	3,538
	Collective, each main engine	3,538	3,545
	Total main engines and air pumps	7,083	
Cooling surface (2) main condensers, square feet		4,500	4,500
	per I.H.P. main engines, square feet	1.272	1.270
Heating surface in use, square feet		13,156	
Grate surface in use, square feet		468	
Total H.S. per total I.H.P., square feet		1.86	
I.H.P. per square foot grate surface, total		15.13	
	ton machinery (penalty weight)	——	
Kind of coal		Welsh, best hand picked.	

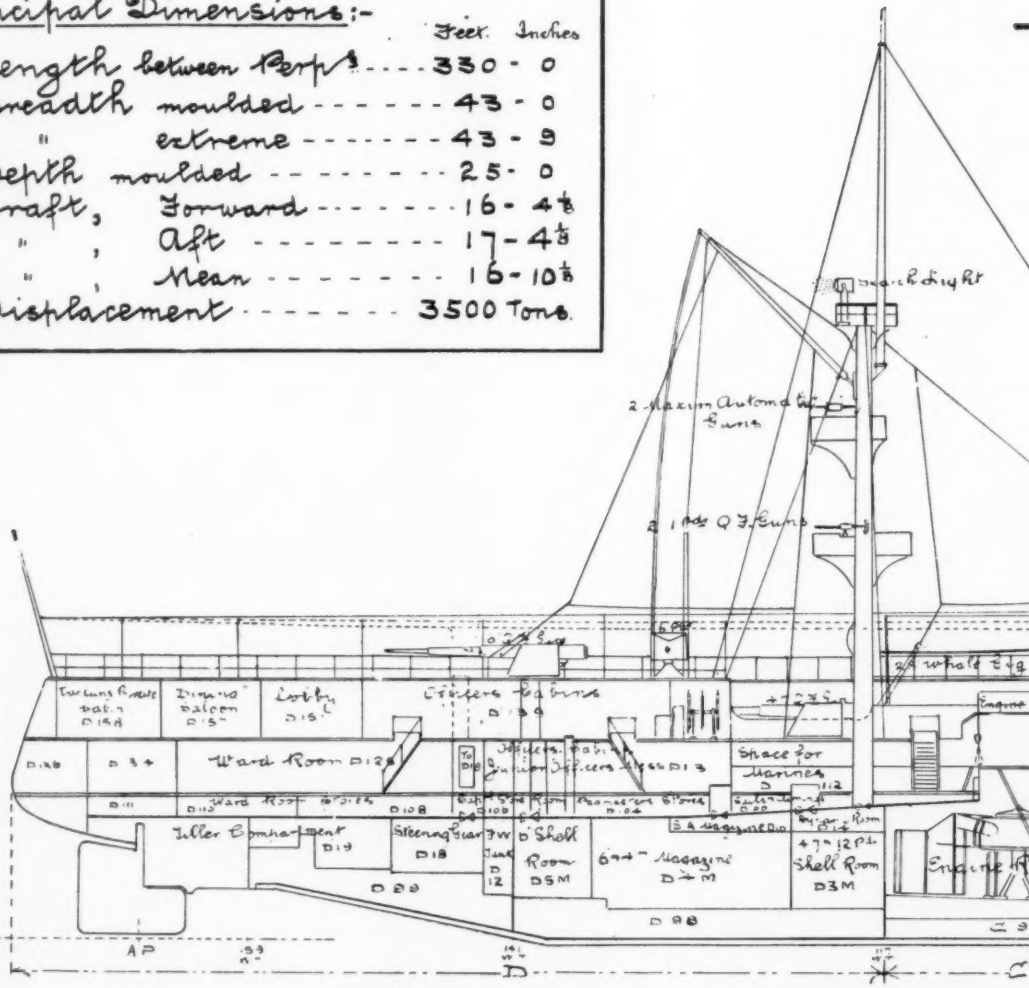
NATURAL-DRAFT DATA—MEAN FOR SIX HOURS' TRIAL.

Steam pressure at boilers (per gage), pounds	148.5
H.P. steam chest, starboard engine (per gage), pounds	131.5
port engine (per gage), pounds	128.3
1st receiver, starboad engine, absolute, pounds	59.15

Steam pressure at 1st receiver, port engine, absolute, pounds..... 58.5
2d receiver, starboard engine, absolute, pounds..... 21.5
port engine, absolute, pounds..... 20.5
Vacuum in condensers in inches of mercury.....port, 25.9; starboard, 27.8
Mean effective pressures in cylinders in pounds per square inch:
Main engines: starboard.....H.P., 49.5; 1st I.P., 27.8; L.P, 9.8
port.....H.P., 51.0; 1st I.P., 26.6; L.P, 8.83
Mean pressure in pounds per square inch on L.P. piston, equivalent to aggregate M.E.P. on all pistons, starboard..... 31.32
Mean pressure in pounds per square inch on L.P. piston, equivalent to aggregate M.E.P. on all pistons, port..... 30.13
Revolutions per minute:
Main engines.....starboard, 148.2; port, 150.6
Pumps, air.....starboard, 148.2; port, 150.6
Pumps, circulating.....starboard, 170; port, 210.0
Pumps, feed, double strokes per minute..... 50.9
Blower engines, forward boiler compartment..... 157
after boiler compartment..... 191
Speed of ship in knots per hour..... 19.52
Slip of propeller in per cent. of its own speed, based on mean pitch, starboard, 15.24; port, 16.58
Air pressure in fire rooms, or in main duct at blower, in inches of water:
Forward boiler compartment..... .22
After boiler compartment..... .25
Indicated horsepower:
Main engines: starboard.....H.P., 826; 1st I.P., 1,028; L.P., 842; total, 2,696
port.....H.P., 865; 1st I.P., 1,000; L.P., 773; total, 2,638

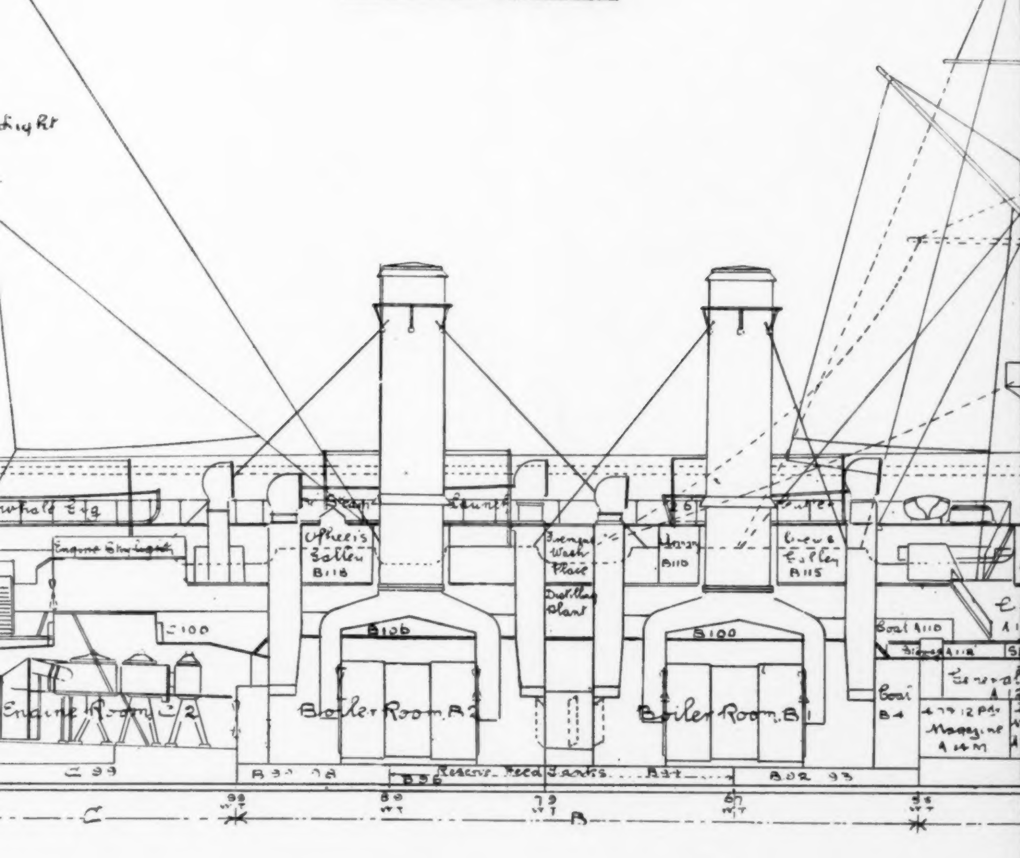
Principal Dimensions:-

	Feet.	Inches
Length between Perps	330	0
Breadth moulded	43	0
" extreme	43	9
Depth moulded	25	0
Draft, Forward	16	4⅛
" , Aft	17	4⅛
" , Mean	16	10⅛
Displacement	3500	Tons.



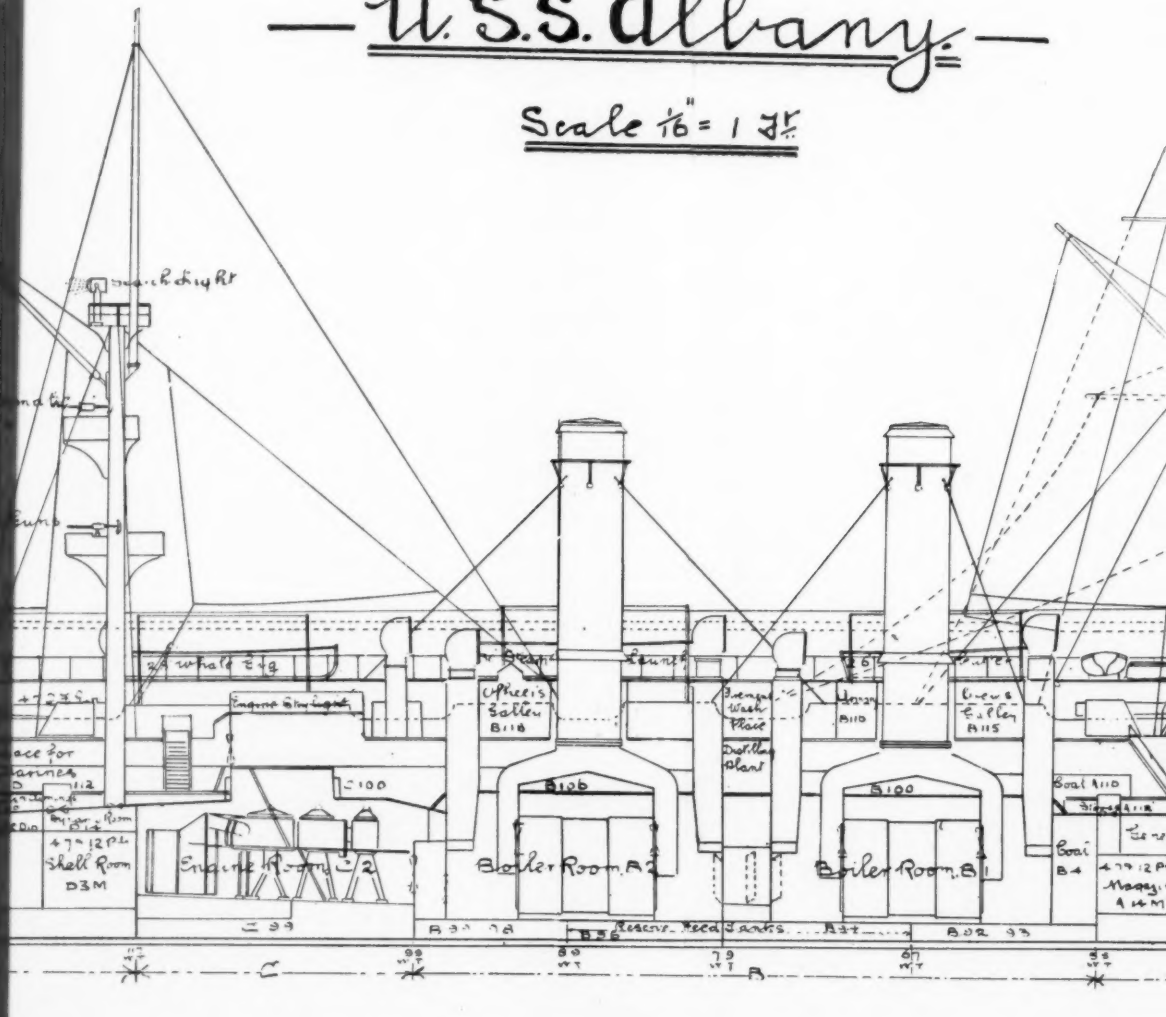
— U.S.S. Albany —

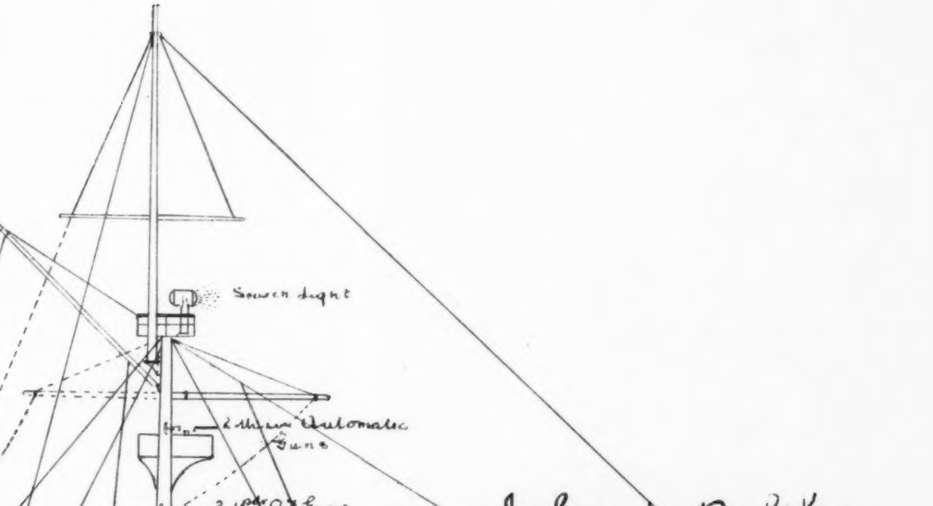
Scale 1/16" = 1 Ft.



U.S.S. Albany

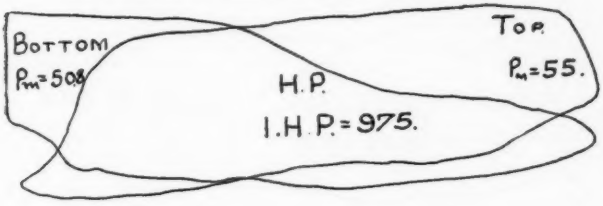
Scale 1/16" = 1 ft

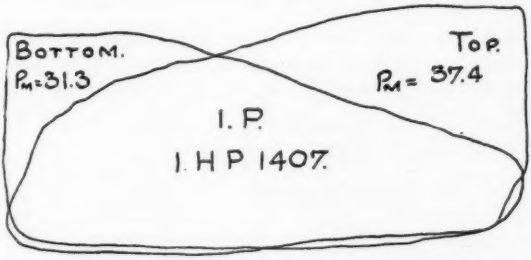


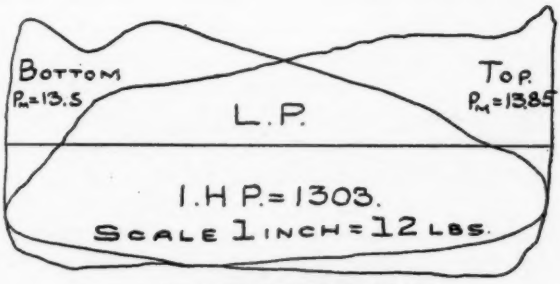


Search Light
Automatic Guns
2 1 Pdr Q.F. Guns
Chart House
Inboard Profile
Anchor Davit
Crew's Quarters
A 132
Crew's
A 122
Quarters
A 121
A 120
A 117
Chain Locker
Stores
A 100
General Stores
6" Shell Room
6" Magazine
Shot Room
A 3
Trimming Tank
A 2
Trimming
Tank
A 99
A 98
A

U. S. S. *ALBANY*. FULL-POWER TRIAL, JANUARY 9, 1900.



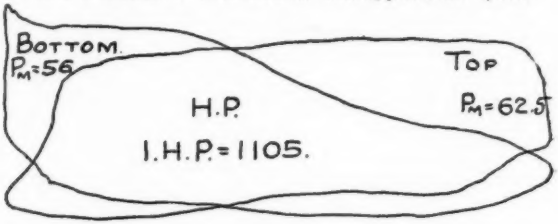




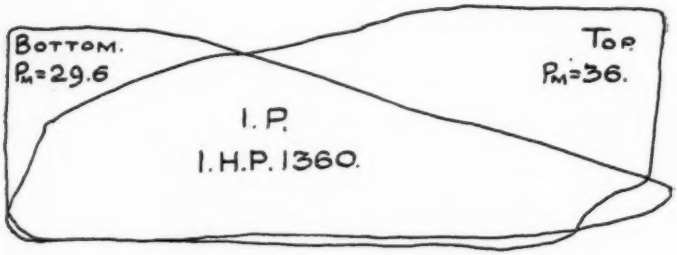
STARBOARD ENGINE-CARDS, NO. 8.

Steam pressure at engine per gage, pounds	140
first receiver, absolute, pounds	69
second receiver, absolute, pounds	26
Revolutions per minute	164
Vacuum, inches	27

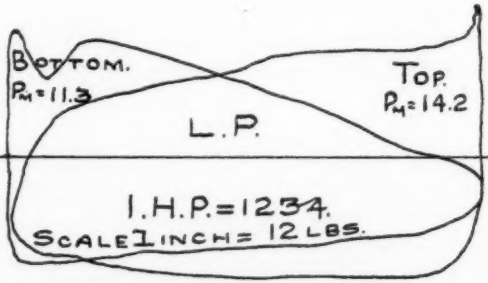
U. S. S. *ALBANY*. FULL-POWER TRIAL, JANUARY 9, 1900.



Scale 1 inch=72 lbs



Scale 1 inch=32. lbs.



PORT ENGINE-CARDS, NO. 8.

Steam pressure at engine per gage, pounds	141
first receiver, absolute, pounds	70
second receiver, absolute	26
Revolutions per minute	166
Vacuum, inches	24.7

THE INFLUENCE OF COMPRESSION IN CYLINDER CLEARANCE SPACES UPON STEAM CONSUMPTION PER I. H. P.*

The elaborate experiments conducted in 1897–8 by Professor Dwelshauvers-Dery of the University of Liège, to ascertain the influence of compression of steam in cylinder dead spaces, excited much notice and comment in the engineering world at the time, and an account of the first series of these experiments was reproduced in the JOURNAL of August, 1898, p. 641.

Among the engineers who took more than an ordinary interest in the Professor's statements it appears that Chief Engineer B. F. Isherwood, U. S. N., followed the experiments most closely, and, while agreeing as to their accuracy, pointed out the true interpretation of the results. This differed from the conclusions of Professor Dwelshauvers-Dery, yet was most gratifying to him inasmuch as it was so convincingly clear. Indeed it induced him to make another set of experiments to verify Mr. Isherwood's deductions. The results of these latter tests are here given as translated from the "Revue Mechanique", prefaced by extracts from the letters from Mr. Isherwood which Professor Dwelshauvers-Dery refers to. These extracts are most interesting and contain the gist of the whole matter.

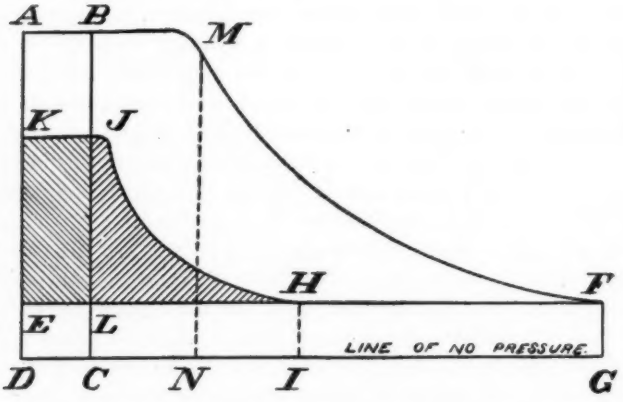
Mr. Isherwood's views are wholly original and are set forth in that lucid and painstaking style so characteristic of his writings, and so conducive to a ready comprehension of the subject on the part of the reader.

The great importance of this particular subject and the notable verification of Mr. Isherwood's theory by this last set of experiments by Professor Dwelshauvers-Dery combine to make a valuable addition to the JOURNAL's pages.

*Extracts from notes by B. F. Isherwood U. S. N. and Prof. Dwelshauvers-Dery.

EXTRACTS FROM A LETTER FROM B. F. ISHERWOOD, U. S. NAVY, TO PROF. DWELSHAUVERS-DERY, JANUARY 30, 1899, ON THE COMPRESSION OF STEAM IN THE DEAD SPACES OF CYLINDERS.

I believe that the following diagram will give you a better idea of my opinions on compression than all I have been able to write to you. The deductions from this diagram apply equally well to every degree of back pressure against the piston, and to every measure of expansion that may be employed. The steam valve is supposed not to have any lead.



In the diagram referred to the rectangle *ABCD* represents the dead space at one end of the cylinder. The height *AD* represents the initial pressure of the steam above the absolute zero. The admission of steam is cut off at the point *M*, from which point the steam expands continuously to the pressure *FG* at the end of the stroke of the piston. The line *EF* represents the back pressure against the piston. During the progress of the piston from *FG* to *HI*, the communication between the cylinder and the condenser remains open; but, at *HI* this communication is closed, and the piston during its farther progress from *HI* to *LC*, the latter being its initial position, compresses the back pressure, the compression pressures forming the curve *HJ*. At the point *J* the compression ceases, although the piston

has not reached the end of its exhaust stroke, because the temperature of the compressed steam is, at *J*, exactly equal to the temperature of the metal of the cylinder upon which it is compressed. Consequently, the compression curve cannot rise above the point *J*, let the distance from *FG* at which the compression commences be what it may. Further, it is physically impossible to compress the back pressure in the cylinder to a pressure equal to the initial pressure in the cylinder, be the back pressure and the degree of compression what they may. The crochet or hook at *J* is found on all indicator diagrams when the steam valve is set without steam lead. When the valve is set with steam lead the crochet is masked or obliterated by the steam entering the cylinder before the piston reaches the end of its exhaust stroke. The boiler steam, on entering, raises the temperature of the metal of the cylinder up to its own temperature by its liquefaction. The hatched surface (////) *JHL* represents the work done by the piston on the pack pressure during the compression; and the hatched surface (\\\\) *JKEL* represents the quantity of steam retained in the cylinder by the compression.

Now this weight of steam retained in the cylinder by the compression is the equivalent of the work *JHL*. Consequently the compressed steam is potentially exactly equivalent to the work expended in its compression, and will exactly reproduce that work during the succeeding stroke of the piston. Hence, under all circumstances, there economically results neither loss nor gain from the compression. The cycle is a closed one; the piston, during the latter part of one stroke, compresses the steam which remains in the cylinder, and this compressed steam reproduces, during the immediately succeeding stroke of the piston exactly the work of its compression. The compression is done by the work of the engine. The compressed steam is just so much added to the boiler steam, and produces a corresponding equivalent of engine work. Steam is a perfectly elastic substance. During the compression *HJ* there is no liquefaction of the compressed steam until the pressure *J* is obtained, the temperature of the metal of the cylinder in contact with the compressed steam being, up to that moment, higher than the temperature of the

compressed steam. From the commencement of the compression at *HI* the compressed steam becomes more and more superheated. The back pressure at the point *H* has a higher temperature than that which is normal to its pressure as saturated steam. The total work done by the steam is represented by the area *BMFGCB*. To do this work there has been drawn from the boiler a quantity of steam represented by the area *ABMNCLEA* less the steam represented by the area *KJLEK*; and this quantity of steam represented by the area *KJLEK* must be added to obtain the quantity used to do the work represented by the area *BMFGCB*. The work done by only the steam drawn from the boiler is represented by the area *BMFHJB* plus the area *FGCLF*, which, necessarily, leaves the work of compression *JHLJ* to be done by the steam represented by the area *KJLEK*. The steam, represented by the area *ELCDE*, remains permanently in the cylinder. It is never exhausted into the condenser since the piston does not move from *LC* to *ED*. Therefore, as taking the steam into consideration would be useless, it can be eliminated from the problem.

The overcoming of the back pressure against the piston, represented by the area *LFGCL*, is as much external work done as the rest of the external work done by the steam, the line *DG* of no pressure should be taken as the base, just the same as in the case of researches on heat, the absolute zero of temperature should be the point of departure. The indicated work is only a fraction, and a very variable one of the total work done.

In experimentally investigating the economic effect of compressing or "cushioning" the back pressure steam, the total work above the line *DG* represented by the area *BMFGCB* minus the work of compression represented by the area *JHL*, should be divided by the weight of steam drawn from the boiler in the same time, or, more precisely, by the corresponding units of heat contained in such steam; when this is done there will be found that the more or less compression has not in the least affected the economy in question. Care must be taken that the reciprocating speed of the piston, and the mean total pressure of the steam on the piston, and the back pressure against the piston, remains the

same. The equality of the piston speed is necessary to equalize the effect of leakage past the piston and valves of the cylinder, but for no other reason. With perfectly tight piston and valves, variation of the piston speed would have no notable influence. The mean total pressure on, and the back pressure *FG* against the piston, should be constant; because variations in them affect the percentage of steam liquefied in the cylinder by the interaction of its metal with the heat of the steam.

If the indicated horses power developed by the engine be taken as the term of comparison, then the same mean indicated pressure must be obtained in the different experiments, and the same back pressure *FG;* also the same speed of piston must be maintained. The reason is that the area *JHLJ* diminishes the indicated area obtained without compression, in consequence of which the back presure area *LFGC* becomes a larger fraction of the total work as the work of compression is increased. With the same initial pressure, the same steam admission, and the same back pressure, compression produces the same economic effect as would be produced without compression by a reduction in the initial pressure.

There is a practical advantage in not compressing the back pressure, due to the smaller dimensions of cylinder then required for the development of the same indicated horses-power. But the dimensions of no other part of the engine need be increased. When compression is employed the cylinder must have more capacity than when compression is not employed in the proportion of the area *BMFLB* on the diagram to that area less the area *JHL*. On the other hand, there is a great practical advantage by "cushioning" in preventing concussion of the moving parts of the engine at the end of each stroke of its piston. The disadvantage and the advantage are, however, purely mechanical, and irrespective of the economic result.

In another letter to Professor Dwelshauvers-Dery, dated the 10th of March following, Mr. Isherwood said:

I here add two deductions from the diagram:

1st. No matter what may be the degree of expansion and of compression adopted, and no matter what may be the proportion

of the dead space at one end of the cylinder to the capacity of the cylinder, there is neither economic gain nor loss by compression of the back pressure. In all cases, the quantity of steam compressed reproduces during the immediately succeeding stroke of the piston exactly the work of that compression.

2d. Compression does not modify in the slightest degree the injurious effect of the dead space on the economy of the steam. The injurious effect of that space remains exactly the same whether there be or be not compression or what may be the degree of compression.

The compression adds just that much steam to the steam generated in the boiler. The compression steam can be considered as so much steam pushed en bloc into the boiler and mixed with the steam already there, although the mixing is, in reality, done in the cylinder. Hence, the weight of compressed steam develops on the piston the same work which the same weight of steam generated in the boiler would develop, and it is this work which is used to compress the back pressure during the succeeding stroke of the piston. Nothing is either gained or lost.

The total power developed by the engine is that which is calculated above the line of no pressure on the diagram; this is not the indicated power which is only a fraction and a very variable fraction of the former. When the comparison is made on this basis, your experiments show that the horsepower is obtained for the same weight of steam per unit of time whatever may be the degree of compression. And if the indicated horses power had, in your experiments, been made equal to the same fraction of the total horses power, those experiments would have given the same economic result, proving the equality of the consumption of steam to power developed be the degree of compression what it might.

RESULTS OF EXPERIMENTS ON COMPRESSION IN STEAM ENGINES, BY PROF. DWELSHAUVERS-DERY, UNIVERSITY OF LIÈGE, 1899.

Mr. Isherwood, the celebrated experimentalist, is the only one who has had a clear insight into the trials that were made in our

laboratory in 1897 and 1898. (See a letter from Mr. Isherwood, "Revue Mechanique," VI, 396). Without questioning our results priori, he has at once made a detailed and complete inquiries into our operations,

I made it my duty to give to him upon his request all the documents relating to our trials, from the first observations to the last calculations, original diagrams included.

Having examined these data minutely he has declared them, as well the results, irreproachable. But to these results he has given an interpretation differing from ours. Without denying the influence of the metal, he thinks that we must not attribute to it so great an influence on the steam consumption, whether compression is used or not; that the difference is due to the increase of the negative work, consequent upon the compression; whereas the positive work, or as Hirn calls it, the absolute work, does not vary, or if so but very little, while the steam consumption is nearly proportioned to the absolute work. He adds: "If my argument is correct, trials similar to yours but made with the absolute work variable with the degree of compression, in such a manner that the indicated work remains the same, should give the same consumption of steam, whatever the degree of compression might be."

This is what I wished to verify by my trials of 1899, which have affirmed the assertions of Mr. Isherwood, the tests of 1897–8 not having lost any of their value as original documents.

In the months of March and April, 1899, we made two sets of experiments, the first being 10 tests without compression, the second 10 with $\frac{4}{10}$ compression. In the following May we then made three tests with $\frac{6}{10}$ compression, and we had intended to continue these, when an accident to the masonry of our superheater interrupted them. At the request of some of our students, we then made two tests at $\frac{6}{10}$ compression, but without superheater.

We have done our utmost that all the trials should be made under the same conditions of indicated work, of speed, of brake load, only varying the pressure in the boiler in one series from the other. This was found to be about as follows:

Without compression about			52,500	kg. p. sq. m.,	74.8	lbs. p. sq. in.
With	4/10	" "	55,500	"	78.9	"
"	6/10	" "	61,500	"	87.4	"
"	6/10	" but without superheater,	62,500	"	89.8	"

The brake load remained constant at 92 kilograms (203 pounds). We sought to obtain a speed of about 47 revolutions per minute. The governor, well extended, was not operative, and the throttle was full open. But from the greatest to the smallest speed we were about 10 per cent. out of the way for a mean speed of 48 revolutions per minute. Since the governor did not operate, we had great difficulty to determine and regulate the brake pressure with a view of obtaining a given speed during the tests. It required a run of two to three hours before the conditions of superheat were assured, and during this time the pressure in the superheater would rise, influencing the speed greatly.

The indicated work, from the smallest to the greatest amount, varied 4.25 per cent. from the mean.

We give with this the figures collected during our experiments.

For convenience of tabulation we have classified the trials in each group in the decreasing order of the indicated horses power. We have added to these the consumption per absolute horses power (Column 4) equal in kilograms of steam, at 655.062 calories per kilogram, and in keeping account of the temperature of the water of circulation from the condenser at its discharge, as well as that which served as feed water to the boiler.

Each trial was at once subjected to the Hirn analysis before being studied. We see by the table (column 5) that the maximum deviation was $+0.95\%$ and inversely -0.31%, the mean for the variation being

$+0.07\%$ for the trials without compression.
$+0.21\%$ for the trials at 4/10 compression.
$+0.73\%$ for the trials at 6/10 compression.

For the trials at 6/10 compression and without superheat the Hirn analysis can not be used as a control, but it may serve to determine the quality of the steam at its entrance into the engine, after the steam chest was cleared. We found this value, as may be seen from the table, to have been a mean of $x=0.995$, the two trials having given respectively 0.998 and 0.992.

The conclusion which evolves directly from this table confirms the views of Mr. Isherwood: *The steam consumption per indicated horses power is practically the same, whatever the compression, as long as the indicated work remains the same.* Actually the mean consumption per I.H.P. was

Without compression 11.116 kgr. (24.55 lbs.).
With 4/10 compression 11.127 kgr. (24.60 lbs.).
With 6/10 compression 11,273 kgr. (24.92 lbs.).

TABLE OF MEAN VALUES.

	Steam consumption.	Compression.			
		None.	4/10	6/10	6/10 Without Superheat.
3	Per H.P., indicated, kilograms...	11.116	11.127	11.213	13.378
3	Per H.P., indicated, pounds.....	24.50	24.55	24.72	29.5
4	Per H.P., absolute, kilograms...	9.648	9.043	7.998	8.794
4	Per H.P., absolute, pounds.....	21.28	19.95	17.65	19.40
5	Hirn's equivalent, per cent.....	+0.07	+0.21	+0.73	$x=0.995$
6	Equivalent in absolute work.....	1.8018	1.9180	2.1915	2.3349
7	Equivalent in indicated work...	1.5653	1.5590	1.5632	1.5352
	Ratio of these equivalents to the total heat Q expended.				
8	Absolute work.....	0.09686	0.10386	0.11730	0.10630
9	Indicated work.....	0.08415	0.08443	0.08365	0.06998
10	Negative work.....	0.01271	0.01943	0.03365	0.03642

In the first series of ten tests without compression, the steam consumption per I.H.P. varied between 10.712 kilograms and 11.526 kilograms (23.60 pounds to 25.38 pounds), a difference of 7.32 per cent., quite an amount. We have not been able to discover the reason thereof, because without doubt the causes

are very complicated, and because the lubrication of the cylinder walls play an important part, the full influence of which it is impossible for us to appreciate.

In the second series of ten tests at $\frac{4}{10}$ compression, the variation was from 10.799 kilograms, to 11.513 kilograms, 23.78 pounds, to 25.38 pounds, or 6.42 per cent. of the mean.

In the third series of three tests at $\frac{6}{10}$ compression, it went from 11.091 kilograms to 11.367 kilograms (24.42 pounds to 25.02 pounds), representing 2.40 per cent. of the mean.

In regard to the consumption per horses power absolute, it was a descending progression to the extent that the pressure in the boiler increased, and the variations of the consumption followed nearly the same law as those which referred to the I.H.P.

	Compression.		
	None.	4/10	6/10
Mean pressure in the boiler, kilogram	52.750	54.900	61.800
Mean pressure in the boiler, pounds	74.9	77.8	87.8
Steam consumption per H.P., absolute, kilograms	9.648	9.043	7.998
Steam consumption per H.P., absolute, pounds	21.25	19.93	17.81
Variation in per cent	7.82	5.92	4.100

It is then that with increasing pressure, the absolute work also increases, and more rapidly than the consumption of heat necessary for doing this work. This follows clearly from columns 6 and 8 of the table. The sixth column gives the equivalent of work in calories, and column 8 the ratio of this quantity to the quantity of Q calories that has been expended in the engine, on the supposition that the boiler had been fed with water of 0 degrees in all the trials.

	Compression.		
	None.	4/10	6/10
The equivalent of work absolute in calories is	1.8018	1.9180	2.1915
And its ratio to the total heat Q expended, in per cent	9.69	10.39	11.30

As regards the production of indicated work, that is, the ratio of the equivalent of indicated work to the total heat expended, Q, it is nearly the same for the three conditions;

Without compression 8.42 per cent.
With 4/10 compression 8.44 per cent.
With 6/10 compression 8.37 per cent.

From the largest to the smallest, the difference is not quite 1 per cent. of the mean. It is this which confirms the prediction of Mr. Isherwood.

A NOVEL METHOD OF INSTALLING PROPELLER SHAFTING.

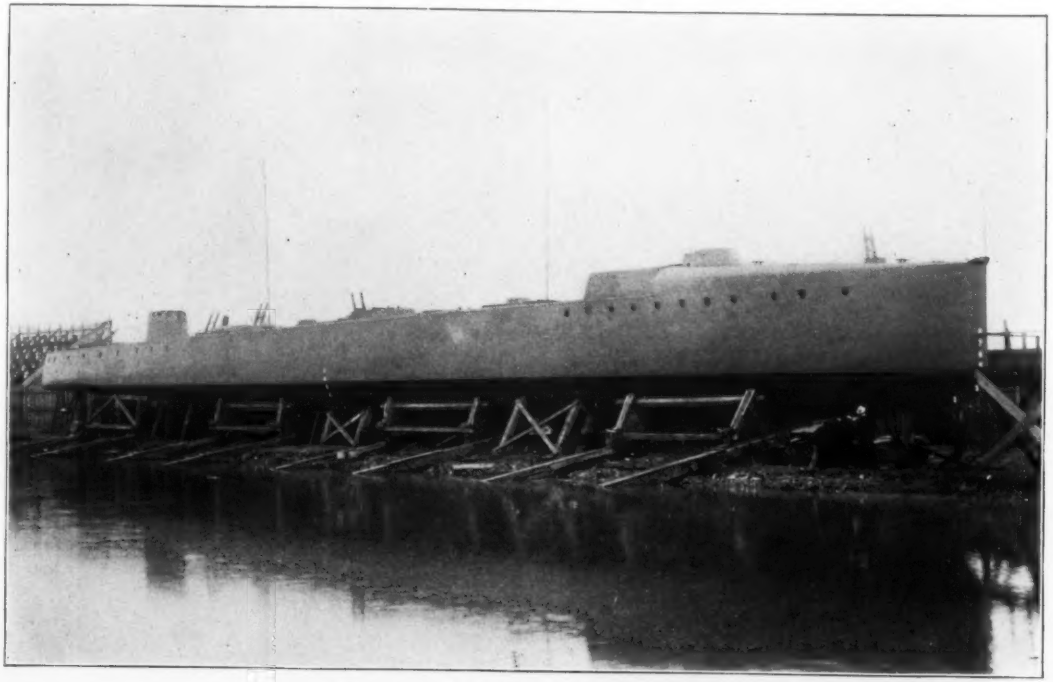
By Commander Harrie Webster, U. S. N.

The torpedo boat *Shubrick* was launched by the builders, the Wm. R. Trigg Company, Richmond, Va., before her propeller shafting had been installed, for several reasons, the principal of which being the otherwise readiness of the vessel long before the contractors for this shafting were able to make delivery, and the desirability of clearing the ways as soon as possible.

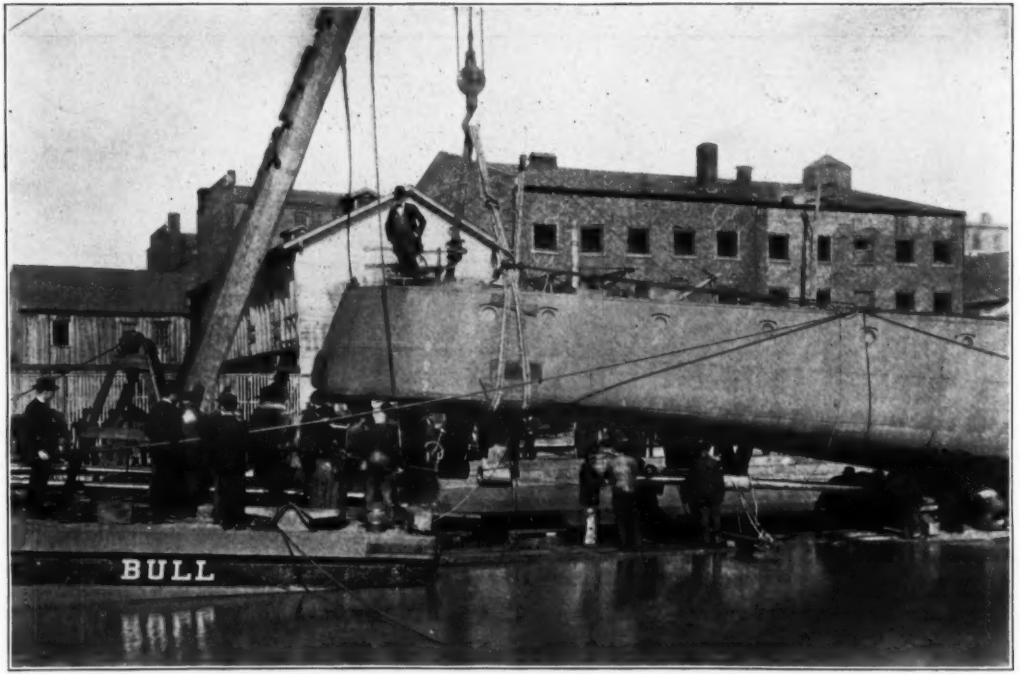
The launch was made on the last day of October, 1899, but it was March, 1900, before the propeller shafting was ready to be put in place, and as Richmond is some hundred miles or so from the nearest dry dock, it became a question of considerable moment touching time and expense, as to whether the long trip and the cost of docking could not be obviated.

This question was promptly solved by the builders' determination to simply raise the stern of the *Shubrick* sufficiently free from the water to enable the work to be done from floats conveniently placed; and the complete success with which the scheme was carried out, together with the slight comparative cost, makes the event well worthy of special notice by shipbuilding and engineering experts.

Commonly speaking, the steel ship, large or small, is supposed to be in the condition of a single element of a bridge structure, and consequently possessing the ability to withstand suspension at any point within the vertical through the center of gravity without distortion or danger. This, however, like many another general hypothesis, is only true within very small limits, and if the attempt were made to suspend a steel craft of any considerable dimensions from a point in the keel directly beneath the center of gravity of the vessel considered as a unit, the result



U. S. TORPEDO BOAT "SHUBRICK" ON THE WAYS.



U. S. TORPEDO BOAT "SHUBRICK" INSTALLING PROPELLER SHAFTING.

would be disastrous. The normal strains to which a floating structure is to be subjected are, as is of course well known, those incident to perfect equability of stress over a large percentage of the exterior area of the floating body, and in addition to its equability is the further advantage of the perfect mobility of the liquid in which the vessel is held in suspension.

So far as it is a matter of record, the raising of a torpedo boat from the water for the purpose of carrying on work actually beneath the vessel is an operation which has not been attempted in this country without the aid of a dry dock.

The accompanying photographs "speak for themselves," and render any written description of the method employed wholly unnecesary. Some details, however, of the derrick, weights, etc., and cost of the operation are added as data of value.

It is also noted that the bow of the *Shubrick* was depressed by the inletting of about 22 tons of water in the forward compartment, facilitating the immersion of the stem and adding to lateral stability during the suspension.

The smooth water of the canal also was a favorable condition, but even with security from disturbance on the part of this element, the daring feature of the attempt remains as an illustration of the confidence the builders held in the excellence of the hull work, especially when it is remembered that the hull plating of a torpedo boat is but about $\frac{1}{8}$ inch thick, and that the length of the *Shubrick* is 175 feet.

DIMENSIONS OF THE DERRICK.

Length, feet	53.08
Breadth, feet	31
Depth, feet	6.66
Draught, feet, normal	2
Displacement, at 2 feet draught, tons	92
Displacement, with *Shubrick* hoisted, tons	137
Dead weight on the derrick float with the *Shubrick* hoisted, which includes men, shafts, and all tackle, tons	45

The 22 tons of water which was let into the *Shubrick* was between the stem and bulkhead No. 27.

The approximate length of the *Shubrick* from point of emer-

gence to perpendicular stern line was 62.5 feet, being somewhat more than one-third of her total length.

The expense attending this method of installation of shafting is given by the Superintendent as follows:

15 men from machine department, . .	$70.97
20 men from ship yard, ...	51.35
Use of derrick for 17 hours, ...	110.50
Making a total of	$232.79

A careful examination made during the period of the vessel's suspension failed to show that any undue strain was imposed on the framing or plating, and the fact that the shafting was emplaced without difficulty was *prima facie* evidence that such was the case.

Inspection of the illustrations will show that the angle of the inclination of the scow under the load of the *Shubrick* was brought almost precisely equal to that of the hull of the torpedo boat, thus rendering the installation of the shafts a comparatively simple matter after the stern of the vessel had been raised.

The cost of docking the *Shubrick*, at Norfolk or Newport News, with incidental expenses due to towing, insurance and delay in work, and other expenses, was estimated by the contractors as in the neighborhood of $1,000.

INERTIA STRESS OF ELASTIC GEARS,

BY

J. H. MACALPINE, Esq.,

ASSOCIATE MEMBER.

TABLE OF CONTENTS AND INDEX OF NOTATION.

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DEDICATED

TO

REAR ADMIRAL GEORGE W. MELVILLE,

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ENGINEER-IN-CHIEF U. S. NAVY,
LL.D., M.SC., DR. ENG'G.

GOLD MEDALLIST, BY ACT OF CONGRESS,
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OF ANTHROPOLOGY AND GEOGRAPHY; INSTITUTION
OF NAVAL ARCHITECTS, GREAT BRITAIN; FRANKLIN IN-
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SCIENCES, &C., &C., &C.

IN GRATEFUL RECOLLECTION OF MUCH KINDNESS
AND FRIENDSHIP.

THE AUTHOR.

INERTIA STRESS OF ELASTIC GEARS.

TABLE OF CONTENTS.

CHAPTER I.

INTRODUCTION.

ARTICLE.

1 General problem.
2 Origin of investigation.
3 Presence of harmonics in valve motion.
4 Inertia stress from valve diagrams. Importance of harmonics.
5 Simple harmonic motion.
6 Rigid and elastic gear.
7 Course of investigation

CHAPTER II.

DETERMINATION OF INERTIA STRESSES FROM EXACT POLAR DIAGRAM—RIGID GEAR. HARMONIC ANALYSIS.

8 Solution of problem.
9 Modifications of method.
10 Numerical examples.
11 Mr. J. Macfarlane Gray's graphical method.
12 Harmonic analysis of two movements.

CHAPTER III.

MOTION OF VALVE WITH ELASTIC GEAR BUT NO FRICTION OR OTHER RESISTANCE.

13 Solution of simplified problem.
14 Remarks on solution.
15 Synchronism.

CHAPTER IV.

FRICTION AND VISCOSITY OF METALS.

16 Law of friction of metals.
17 Friction proportional to velocity.
18 Viscosity of metals.

CHAPTER V.

MOTION OF VALVE WITH FRICTION. FRICTION MUCH LARGER THAN RIGID GEAR INERTIA.

19 Change of center of vibration through friction at reversal.
20 Motion when J is large.
21 Numerical results—greatly increased inertia.

CHAPTER VI.

THREE ILLUSTRATIONS OF FRICTIONALLY RESISTED MOTION. ENERGY TRANSFER.

1st Illustration.

22 Mass moving under elasticity of spring alone.
23 Measurements of energy for *1st Illustration.*

2d Illustration.

24 First period and synchronizing vibrations only. Actual epochs each zero.
25 Change of a_s' per revolution. Limiting value of a_s.

3d Illustration.

26 1st period and synchronizing vibrations only. Actual epochs zero and 180°, respectively.
27 Increase of amplitude per revolution.
28 Possible large increase of inertia. Stability of epoch.
29 Curves of energy transfer at reversal explained. Transfers in *2d and 3d Illustrations.* Multiple reversals.

CHAPTER VII.

GENERAL THEORY OF MOTION RESISTED BY CONSTANT FRICTION.

30 General change of motion at reversal.
31 The motion and changes at reversal represented graphically.

1st Proposition.

32 Conditions of stability of epoch of actual synchronizing vibration, *3d Illustration.*

2d Proposition.

33 Conditions of stability of epoch of actual synchronizing vibration, *2d Illustration.*
34 Definitions of *Normal Epoch, Ordinate and Vector.*
35 Two conditions that the events in successive revolutions may be the same.—1st and synchronizing periods alone present.
36 In simple case, § 35, ε_s' must change towards 180° till triple reversals occur.
37 I_s may become important though $a_s = 0$.
38 Two conditions that events in successive revolutions may be the same. Actual case.
39 Changes of a_s' and ε_s' small, per revolution if J and J' are small.

CHAPTER VIII.

MOTION RESISTED BY CONSTANT FRICTION. J AND J' SMALL.

40 Curves of change of ε_s' at reversal.
41 Maximum change of ε_s' by one reversal.
42 Rate of restoration of normal epoch by growth of a_s'.
43 Determination and curves of energy transfer due to friction at reversal.
44 Energy supplied by the gear to the actual synchronizing vibration, per revolution.
45 Expression of *2d Condition*, § 38.

CHAPTER IX.

CURVES OF ENERGY TRANSFER AND CHANGE OF EPOCH APPLIED.

46 Actual conditions—difficulty of some cases.
47 Data required, s, I_s and I_f.
48 Method of obtaining energy transfer curve due to friction.
49 Example from *Newark.*

50 Method of obtaining curve of change of epoch due to friction.
51 Curve of restoration of normal epoch, and example from *Newark*.
52 Expression of *1st Condition*, § 38.

CHAPTER X.

HARMONIC ANALYSIS OF MARSHALL GEAR.

53 Simple approximate law of higher period amplitudes, equations (12) to (15).
54 Additional results like equations (12) to (15).
55 Substituted geometrical series.
56 Division of total valve movement.
57 Equivalent simple gear.
58 Parts of movement unimportant for present investigation.
59 Remarks on inaccuracies in analysis.
60 Epoch of high period rigid gear vibrations.

CHAPTER XI.

APPLICATION OF PRECEDING THEORIES TO U. S. S. *NEWARK'S* L.P. VALVE GEAR.

61 Estimation of a_6.
62 Calculation of the elasticity of the gear.
63 Remarks on *Table VIII.* Advantage of Stephenson's gear. Dr. Kirk's experience.
64 Calculation of natural period and s.
65 Data from Official Reports.
66 Estimate of probable value of F.
67 Determination of energy transfer curves and normal epoch.
68 Determination of F for $n_e = 2.0$ — full ahead gear.
69 Stress in studs which broke.
70 Intensity of stress which would probably determine fracture.
71 Calculation of stress for .76, .725 and .65 cut off—*Table X.*
72 Remarks on results in *Table X.*
73 Sharpness of determination of safe angle of reversing lever.
74 Official Reports. Entire agreement with results in *Table X.*
75 Great increase of a_s' over a_s.

CHAPTER XII.

U. S. S. *CHICAGO'S* VALVE GEAR. EFFECT OF STIFFENING GEARS. SIMILAR GEARS.

76 Change in *Chicago's* gear—consequent 1st period movement.
77 6th period movement, rigid gear.
78 Comparison of valve motions of *Chicago* and *Newark*.
79 Remarks on various gears, U. S. S. *Katahdin* and *Petrel*.
80 Effect of stiffening gears. General considerations.
81 Solution when stiffening raises synchronizing term from s to $s + 1$.
82 Similar gears having s and F/I_e the same will be similarly stressed.

CHAPTER XIII.

FRICTION PROPORTIONAL TO THE VELOCITY. VISCOSITY.

83 Friction proportional to the velocity.

Viscosity.

84 Nature of conditions and stresses in a machine. Effect on coefficient of viscosity.
85 Absorption of energy by friction and viscosity compared.
86 Value of a_s'/a_s when viscosity alone acts.
87 Required stiffness of gear if viscosity alone acts. *Newark* and *Chicago*.
88 Effect of stiffening gears.
89 Small effect of viscosity in previous calculations regarding *Newark*.
90 Similar gears similarly stressed.

CHAPTER XIV.

FACTORS OF SAFETY. CONCLUSION.

91 Frequent occurrence of high inertia stress.
92 Wöhler's experiments.
93 Ordinary Factors of Safety and necessary factors from Wöhler's experiments compared.

94 Reasons from preceding investigation for high factors of safety.
95 Factor of Safety for a dead load.
96 Conclusion.

APPENDIX A.

NOTE ON ZEUNER'S DIAGRAM.

APPENDIX B.

EFFECT OF VARIOUS ERRORS ON THE ESTIMATION OF THE RIGID GEAR INERTIA STRESS.

APPENDIX C.

HARMONIC ANALYSIS OF THE MOTION OF THE POINT *A*, FIGS. 4 TO 7.

APPENDIX D.

MOTION OF A MASS SUSPENDED BY A SPRING THE UPPER END OF WHICH HAS ANY ARBITRARY VERTICAL PERIODIC MOTION.

(1) No Resistance to Motion.
(2) Friction Proportional to Velocity.

APPENDIX E.

NATURAL PERIODS OF U. S. S. *NEWARK'S* L. P. VALVES.

INDEX OF NOTATION.

a Amplitude of vertical movement of A, Figs. 4 to 7.

a Radius ES, Figs. 26 and 31.

a_s Amplitude of substituted *sine* curve, *1st Proposition.*

a_1 Amplitude of 1st period, rigid gear.

a_2, a_3, &c. Amplitude of 2d, 3d, &c., period, rigid gear.

a_n Amplitude of nth period, rigid gear.

a_s Amplitude of synchronizing term, rigid gear.

a_{s+1} Amplitude of rigid gear synchronizing vibration when the gear is stiffened so that $s+1$ becomes the number of the synchronizing period.

a_1', a_2', a_3', &c. Amplitude of 1st, 2d, 3d, &c., periods, elastic gear.

a_n' Amplitude of nth period, elastic gear $= a_n \omega_s^2 / (\omega_s^2 - n^2 \omega^2) = a_n / (1 - T^2/T_n^2)$, when there is no resistance or constant resistance F.

a_s' Amplitude of actual synchronizing vibration, elastic gear, $= B_0 + a_s \omega_s t/2$, when there is no resistance or constant resistance F.

b Constant term and amplitude of 2d period, equation (97).

c Constant in formulas A and B; § 16, footnote.

d Sign of differentiation.

e Base of Naperian logarithms, equation (104).

e Proportionate error in measuring *sec* φ, *Appendix B.*

e_1 Proportionate error in measuring ρ, *Appendix B.*

f Elastic strength of gear in pounds. Force $f\delta r$ displaces M by distance δr feet.

f_1 Do. when the gear is stiffened so that $s + 1$ becomes the number of the synchronizing period.

f_1 Elastic strength of valve V_1, Figs. 27 and 28, in pounds.

f_2 Elastic strength of valve V_2, Figs. 27 and 28, in pounds.

g Intensity of gravity = 32.2 feet per second.

j Ratio of substituted geometrical series, marshall gear. Equation (93).

k Constant term in displacement of mass M, rigid gear; and elastic gear, when there is no resistance.

k Mean height of point A, Figs. 4 to 7, = distance of C below OX.

k' Constant term in equation ($4d$), *Appendix D.* $k' = k$.

k_s Constant term in substituted *sine* curve, *1st Proposition.*

m Ratio A_1S/AS, Fig. 26.

n Any whole number. Used in distinguishing periodic displacements, &c.; as "nth period." Also used as a suffix, as a_n.

$n_e = I_s/I_e$ = number of curves, Fig. 21, 16_2, &c., to be used for reversals near e, Fig. 29, any cut off.

$n_f = I_s/I_f$ = number of curves, Fig. 21, 16_2, &c., to be used for reversals near f, Fig. 29, any cut off.

p Perpendicular from center of curvature on radius vector of point, in polar diagram.

p Normal pressure in formulas A and B; § 16, footnote.

r Displacement of M, rigid gear.

r Displacement of upper end of spring suspending mass M.

r Displacement of piston from top stroke, equations (10) and (11).

r Horizontal displacement of A, Figs. 4 to 7, from OY.

r_1 Actual displacement of mass M—sometimes only up to time of reversal of motion.

r_2 Actual displacement of mass M after reversal of motion.

r_1 Total displacement of valve V_1, Figs. 27 and 28, *Appendix E.*

r_2 Total displacement of valve V_2, Figs. 27 and 28, *Appendix E.*

r' $\delta r' = F/f$.

r_1' Displacement of valve V_1, Figs. 27 and 28, due to force applied to it; *Appendix E.*

r_2' Displacement of valve V_2, Figs. 27 and 28, due to force applied to it; *Appendix E.*

r_e Maximum valve displacement near Oe, Fig. 29, at any cut off.

r_f Maximum valve displacement near Of, Fig. 29, at any cut off.

r_s $\delta r_s =$ difference of terms (23) and (24).

s The number of the period of the synchronizing term. Used as a suffix to indicate any function relating to the synchronizing term, as a_s.

t Time in seconds.

t_1 Time when the cycle begins, equation (25). $\omega_s t_1 - \varepsilon_s = 2n\pi$.

t_2 Time eh, Fig. 8, at which mass M restarts.

t_r Time of reversal of motion of mass M.

t' $\omega_s t' = \omega_s t - \varepsilon_s$, a substitution in *Appendix D.*

v Velocity of mass M in feet per second.

v Velocity of sliding in formulas A and B; § 16, footnote.

x x, y, coordinates of B, Fig. 26.

x_1 x_1, y_1, coordinates of B_1, Fig. 26.

y See x.

y Vertical displacement of A, Figs. 4 to 7.

y_1 See x_1.

A Arbitrary amplitude in term of natural period, $A\cos(\omega_n t - \lambda)$, solution of equation ($1d'$), &c.

A_0 Arbitrary amplitude in equation (25_2).

A' Arbitrary constant.

A'' Arbitrary constant.

A_1 Arbitrary amplitude in term of natural period, $A_1\cos(\omega_n t - \lambda_1)$, solution of equation ($1d'$) after reversal.

B_0 Amplitude, when $t = 0$, of that part of the actual synchronizing vibration which has normal epoch.

B' Arbitrary constant.

B'' Arbitrary constant.

$C = A \sin \lambda$.

$E°$ Angle between Oc and the maximum vector near Oe, Fig. 29, at any cut off.

$F°$ Angle between Od and the maximum vector near Of, Fig. 29, at any cut off.

F Friction of mass M, = constant.

F Friction in formulas A and B; § 16, footnote.

I Inertia of mass M, rigid gear.

I_b Inertia of mass M, at b, Fig. 8, rigid gear.

I_e Inertia of valve at maximum displacement near e, Fig. 29, any cut off, rigid gear.

I_f Inertia of valve at maximum displacement near f, Fig. 29, any cut off, rigid gear.

I_e' Inertia of valve at maximum displacement near e, Fig. 29, $A = 0$, any cut off, elastic gear.

I_f' Inertia of valve at maximum displacement near f, Fig. 29, $A = 0$, any cut off, elastic gear.

I' Inertia of mass M when $A = 0$, elastic gear.

I_e' Inertia of mass M at e, Fig. 8, elastic gear.

I_s Maximum inertia of valve from synchronizing vibration, elastic gear, $I_s = fa_s'$ when a_s' may be considered constant and a_s'/a^2 is large.

I_{s+1} Do. when the gear is stiffened so that $s+1$ becomes the number of the synchronizing term.

J $= F/$Rigid gear inertia of valve at reversal $= F/I_e$ or F/I_f.

J' $= F/I_s$.

K Constant of formulas A and B; § 16, footnote.

M Mass of valve in pounds.

M Mass of vibrating mass hung by spring, in pounds.

M_1 Mass of valve V_1, Figs. 27 and 28, with spindle, &c., in pounds.

M_2 Mass of valve V_2, Figs. 27 and 28, with spindle, &c., in pounds.

N_2, N_4, &c. Coefficients in equation (3c) &c., *Appendix C.*

P_0, P_1, P_2, &c. Coefficients in *Table Ic*, &c., *Appendix C.*

Q_0 Constant term of series giving horizontal movement of A, Figs. 4 to 7.

Q_1, Q_2, Q_3, &c. Amplitude of 1st, 2d, 3d, &c., terms of series giving horizontal movement of A, Figs. 4 to 7.

Q_0' First term of substituted geometrical series, Marshall gear.

Q_1', Q_2', Q_3', &c. Successive terms of substituted geometrical series, Marshall gear.

R Radius of crank, equations (10) and (11).

R Radius OA, Figs. 4 to 7, 26 and 31.

R_1 Equivalent radius rod, O_1A_1, Fig. 26.

R_e Revolutions of engine per minute.

S Total stress of valve in pounds.

T Natural period of vibration of mass M in seconds.

T_1 Periodic time, in seconds, of 1st period $=$ time of revolution of engine.

T_2 Periodic time, in seconds, of 2d period $= T_1/2$.

T_3 Periodic time, in seconds, of 3d period $= T_1/3$.

T_n Periodic time, in seconds, of nth period $= T_1/n$.

T_s Periodic time, in seconds, of synchronizing term, $T_s = T$.

V Friction in pounds of mass M when the velocity is one foot per second—friction proportional to first power of velocity, Chapter XIII.

a Difference of true epoch from normal epoch in actual synchronizing vibration. $a = \varepsilon_s' - \varepsilon_s''$.

a_1 Do. when the gear is stiffened so that $s + 1$ becomes the number of the synchronizing term.

β Angle by which the eccentric is in advance of the crank, Marshall gear.

γ Constant in formulas A and B; § 16, footnote.

γ_1 $\gamma_1 r_1' =$ consequent displacement of valve V_2, Figs. 27 and 28, when valve V_1 is forcibly displaced by r_1'.

γ_2 $\gamma_2 r_2' =$ consequent displacement of valve V_1, Figs. 27 and 28, when valve V_2 is forcibly displaced by r_2'.

δ Sign of finite increment, as δr.

ε Small angle in term $-b \sin 2(\theta + \varepsilon)$, equation (97).

$\varepsilon_1, \varepsilon_2, \varepsilon_3$, &c. Epoch of 1st, 2d, 3d, &c., period vibrations respectively, rigid gear.

ε_n Epoch of nth period vibrations, rigid gear.

ε_s Epoch of synchronizing vibrations, rigid gear.

$\varepsilon_1', \varepsilon_2', \varepsilon_3'$, &c. Epoch of 1st, 2d, 3d, &c., period vibrations respectively, elastic gear.

ε_n' Epoch of nth period vibrations, elastic gear.

ε_s' Epoch of synchronizing vibration, elastic gear.

ε_s'' Normal epoch of synchronizing vibration, elastic gear.

η Small angle, ultimately vanishing, equation (10d), &c., *Appendix D.*

θ Angle of crank in radians, measured from SY, Figs. 26 and 31.

κ Coefficient of change of epoch due to friction; left hand scale reading of ordinates of curve *aa*, Figs. 30_1, 30_2, 30_3, 30_4. Change of epoch per revolution of engine due to friction $= 2F\kappa/I_s$ radians.

λ Arbitrary epoch in term of natural period, $A\cos(\omega_s t - \lambda)$, solution of equation (1*d*), &c.

λ_1 Arbitrary epoch in term of natural period, $A_1\cos(\omega_s t\lambda - \lambda_1)$, solution of equation (1*d*), &c., after reversal.

$\mu = V_g/M$.

ν_n Difference of epochs of rigid gear and actual vibrations of nth period, when the friction of the valve is proportional to the velocity.

ν_s Do. for the synchronizing vibrations. $\nu_s = \frac{\pi}{2}$.

ξ Angle LSY_1, Fig. 31.

π Ratio of circumference to diameter of circle.

ρ Radius of curvature in polar valve diagram.

ρ_e Radius of curvature at maximum vector near *Oe*, Fig. 29, at any cut off.

ρ_f Radius of curvature at maximum vector near *Of*, Fig. 29, at any cut off.

τ Coefficient of energy transfer due to friction. $\tau F a_s' =$ rate of absorption of energy, per revolution of engine, by friction F.

τ_1 Do. when the gear is stiffened so that $s + 1$ becomes the number of the synchronizing term.

φ Angle between normal and radius vector at point, polar valve diagram.

$\chi_1 = \sqrt{f_1 g/M_1}$, *Appendix E.*

$\chi_2 = \sqrt{f_2 g/M_2}$, *Appendix E.*

ψ Angle of radius rod OA to XX_1, Fig. 26, when the lower center of the eccentric beam is at S.

ψ_1 Do. for the equivalent radius O_1A_1.

ω Angular velocity of crank in radians per second.

ω_s Angular velocity for substituted *sine* curve, *1st Proposition.*

ω_s Angular velocity for synchronizing term; that is, of an arm revolving in time T. $\omega_s = \sqrt{fg/M}$.

ω_{s+1} Do. when the gear is stiffened so that $s + 1$ becomes the number of the synchronizing term.

Σ Sign of summation.

SPECIAL DEFINITIONS, ETC.

ARTICLE.

4 Positive force on gear. Footnote.

5 Simple Harmonic Motion. Amplitude, Period, Epoch, Phase.

5 1st, 2d, . . . nth, Period.

6 Rigid gear. Actual or elastic gear.

29 Single, triple and multiple reversal.

34 Normal Epoch, Ordinate and Vector.

60 High and low maxima of B_1, Fig. 26. All equations in appendices are marked with a corresponding letter; as equation (10*d*) in *Appendix D.*

INERTIA STRESS OF ELASTIC GEARS.

DEDICATED TO REAR ADMIRAL GEORGE W. MELVILLE, U. S. N.,
BY THE AUTHOR, J. H. MACALPINE, ESQUIRE,
ASSOCIATE MEMBER.

CHAPTER I.

INTRODUCTION.

1. GENERAL PROBLEM.

Every solid possesses some degree of elasticity of shape. Hence the problem here proposed is, in a restricted form, one of the most general problems of structural design.

Each part of every structure or machine is submitted to forces which can rarely be considered quite constant and consequently—between definite time limits—may be treated as periodic. If any elastic body is submitted to periodic forces or movements we know that it will be thrown into vibration and that the period or periods of vibration will be the same as those of the simple harmonic elements into which the total disturbing forces or movements can be analyzed. The induced vibration will reach a definite intensity, producing definite maximum inertia stresses at each point. To calculate those stresses exactly would require complete knowledge of the modes of vibration induced; which, in the large proportion of cases, is a problem of insuperable difficulty.

If there is synchronism between any element of the disturbance and the induced vibration we know that this portion of the vibration will, in a perfectly elastic solid free from any cause of dissipation of energy, go on increasing till the limits of elasticity are passed or rupture is produced.

But, in every actual solid, vibration is accompanied by dissipation of energy, as is at once evident from the fact that the vibration always begins to subside the moment the exciting cause is removed. That part of the dissipation which is due to internal molecular action of the solid is termed *viscosity*. We

have, also, in the friction produced by the sliding of the parts, especially in machines, a most important cause of this dissipation; while the resistance of the medium in which the movement takes place sometimes powerfully, but in most cases feebly, contributes to this effect. Curiously, friction constant in amount, opposing the motion, has, under certain definite conditions, precisely the opposite effect, and supplies energy to the synchronizing vibration till it has reached a definite intensity; and this fact, we will find, leads to results of importance.

The result of the interaction of elasticity and dissipation of energy is of fundamental importance to engineering practice and has never, I believe, been sufficiently discussed.

In elastic gears, such as valve gears actuating a comparatively large mass, we have a simple case in which the natural period and all the particulars of the motion can often be readily ascertained. This investigation arose in connection with some valve gears which developed unexpectedly large stresses; and as the illustrations are nearly all taken from valve gears the mass actuated is usually referred to as the valve.

Without at present remarking further on the larger problem I shall show how that with which the investigation deals directly, presented itself to my mind.

2. ORIGIN OF INVESTIGATION.

Among marine engineers on both sides of the Atlantic, from ten to fifteen years ago, there was a very general movement towards the adoption of non-eccentric and single-eccentric valve gears. This tendency affected to a lesser extent other branches of our profession, and it seemed for a time that the long and remarkable reign of the Stephenson Link Motion was seriously threatened. Principal among these new gears was the Marshall, a modification of that discovered by Hackworth about the same time as Stephenson gave his immortal invention to the world. Joy's first gear also attracted much attention; and there were about half a dozen, lesser known, in some cases old inventions revived, which gained frequent application. The name of the gears invented during this period, whose silent record is to be found in the Patent Office, is legion.

The principal objects aimed at were simplicity of the gear itself and accessibility of the engine. These objects were in some cases attained to a considerable extent, and I believe several of the gears have established for themselves a permanent footing. While admiring the wonderful genius of Stephenson, surely no one can be blind to the great mechanical beauty of the inventions of Hackworth, Marshall, and Hensinger von Waldegg, and I strongly hold the opinion that these gears are esteemed to-day far below their real worth.

There has been a marked reversion towards Stephenson. With some of the newly-adopted gears it was found, especially in comparatively high-speed engines, that the parts were submitted to a much heavier stress than was anticipated. Piston valves had come into favor, as their small friction, it was supposed, would lead not only to diminished wear and tear, but to light stresses and correspondingly light gear. That there was a fallacy somewhere the trials of a considerable proportion of these engines abundantly proved. There were endless breakages of parts which calculation seemed to show were amply strong, and even where the gear remained intact there was often a very noticeable and disconcerting short-period vibration set up.

It must have been very generally appreciated that the large increase of stress and the occurrence of sensible vibration were due to an element in the valve motion synchronizing more or less nearly with the natural period of the valve.

In the case of a weight suspended by a spring we may draw the weight downward and release it with the result that it makes, say, three vibrations per second. If, now, we submit the upper end of the spring to a small vertical displacement, downward and upward, repeated three times per second—or apply, at the same rate, a small vertical force to the suspended weight—we will gradually induce a large vibration, the effect being apparently out of all proportion to the exciting cause. The growing vibration will not be affected if the small displacement, or force, is accompanied, as in the case of a valve, with movements or forces of longer period.

But many questions arise: What are the conditions which

have to be fulfilled that a gear will run steadily, or the elastic structure be apparently quite stationary, so that no abnormal stresses will be induced? If the limits imposed by these conditions are passed, and especially if synchronism is established, to what extent will the amplitude of the vibration and consequent stress grow? Are we sure that, even when the gear is running steadily so far as we can judge, the stresses are not much more intense than is shown by the ordinary methods of calculation? The sequel will show that the answers to these questions involved many others which we need not set down here.

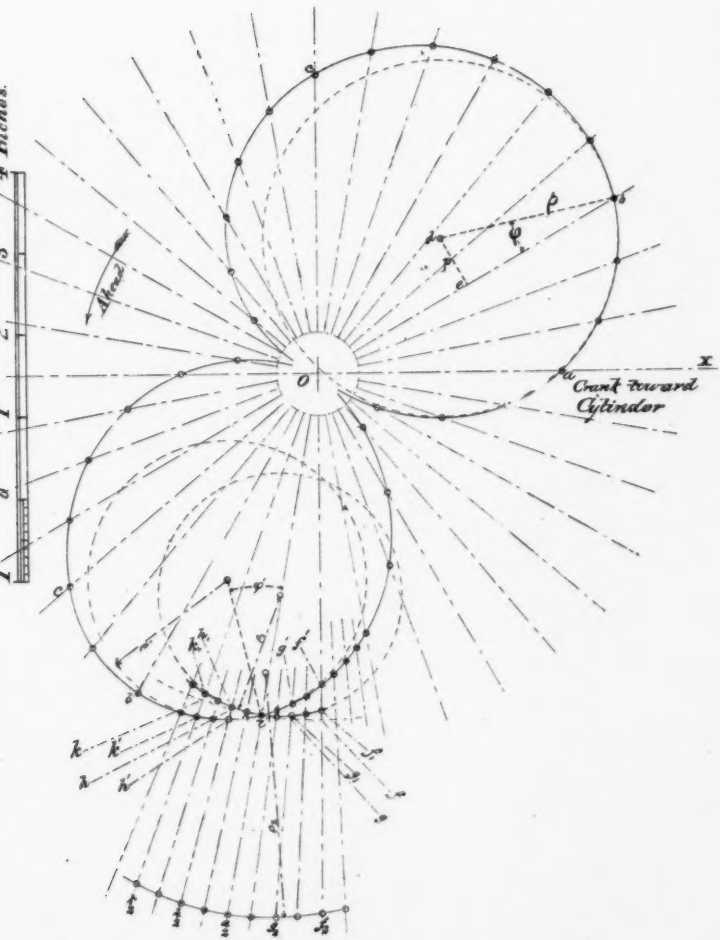
So far as I am aware there has only been one attempt to determine experimentally the stresses induced in the actual working of a valve gear, and this led to no very definite result. It was not to be anticipated that it would, for the maximum intensity of stress is not connected in any simple way with the speed of revolution; that is, it does not rise according to any simple law as the revolutions increase. The conditions are such that it is hardly conceivable that they would be discovered by any practical series of experiments, and any results obtained from one gear would usually appear contradictory to those from another. The problem is purely a dynamical one and is most easily attacked by analysis, the resulting theory being verified by application to the known results of trials. This is the procedure in the present investigation, and I show by reference to unsuccessful and successful trials that it is possible to determine sharply the limiting conditions of success.

My attention was powerfully drawn to the present problem from ten to twelve years ago by the very unsatisfactory working of two valve gears with the designs of which I had been closely connected. In the first, the engine ran most satisfactorily till it reached between 90 and 100 revolutions; above that the low pressure and intermediate valve gears began to work unsteadily, and on some of the trials actually broke down. It was only by altering the gears so as to stiffen them that a satisfactory full-power trial could be obtained. In the second case, though the valve gear was much stronger in proportion than the former, it began to give trouble at even lower revolutions. It was only

XU

4 Inches.
3
2
1
0
1
c
k
Full

Fig. 1.



Full Ahead L.P. Valve Diagram
U.S.S. "Newark."

by again stiffening the gear and linking it in a little from its longest cut off that a fairly satisfactory run was made and a breakdown avoided.

I determined to investigate the question if a suitable opportunity offered, but it was evidently a work of some magnitude. The furthest I could proceed then was to deduce a method of determining the stresses direct from the valve ellipse, which, of course, neglected all effects of elasticity. This I embodied in a paper to which I shall refer immediately. Careful measurements showed that these stresses rose far above that due to simple harmonic motion. While this was a step in the right direction the sequel will show that it was far below the true result. The natural period of the intermediate valve gear in the first of these two engines was also calculated, it being found to be about one-thirteenth of a second; so that, when the engine was running at full speed—110 revolutions per minute—the valve would execute about 7 complete vibrations of natural period per revolution of the engine. It must have been a simple harmonic component of the valve motion, with this period, which gave rise to the synchronous vibration and produced the unsteady working of the valve.

When, in 1892, I had the honor to become acquainted with Admiral (then Commodore) Melville, Engineer-in-Chief of the U. S. Navy, I found that he had had the same trouble in some half dozen cases and had almost determined to abandon the Marshall gear. In the autumn of 1897, at which time I was in the Bureau of Steam Engineering, where he is chief, I requested permission to attack the problem, and it is entirely due to his sympathetic interest and great patience that I have been able to carry it to completion. The work, though very far advanced, was still incomplete when I left America in May, 1898, and the long delay since then has been due to extreme pressure of other engagements.

3. PRESENCE OF HARMONICS IN VALVE MOTION.

That such simple harmonic elements exist in valve motions is known to all readers of Zeuner's Treatise on Valve Gears. In that classical work he clearly points out, and for some valve

gears treated he investigates an approximate expression for, the variation of the valve displacement from his circle diagram. This variation he calls the "missing quantity." These expressions are amply accurate for the purpose he has in view; namely, to prove that, usually, his circle diagram gives very closely the elements on which the distribution of steam in the engine depends. A glance at his investigation shows that the deduced value of the "missing quantity" depends on various approximate assumptions, and on some conditions of the setting of the gear which it is desirable to fulfil but which may have to be departed from to some extent in any particular design.

The exact expression for the valve motion would present itself in the form of an infinite series, thus—

$$r = k + a_1 cos(\theta - \varepsilon_1) + a_2 cos(2\theta - \varepsilon_2) + a_3 cos(3\theta - \varepsilon_3) + \ . \ . \ ; \ . \quad (1)$$

where r is the displacement of the valve from any arbitrarily-chosen position in its line of motion; θ the angle of the crank; $k, a_1, a_2, a_3, \ldots$ are numerical constants rapidly diminishing in value as we proceed along the series; and $\varepsilon_1, \varepsilon_2, \varepsilon_3, \ldots$ constant angles, determinate from the dimensions and setting of the gear.

The exact L.P. Diagram of U. S. S. *Newark*, when full ahead, is shown in Fig. 1, the diagram being a polar one exactly like Zeuner's. O being the center of the crank shaft and Oa, Ob, Oc, ... various positions of the crank, the corresponding displacements of the valve from any arbitrarily-chosen position in the line of its motion are given by the lengths of the vectors $Oa, Ob, Oc, \ldots$ respectively. The result is a pair of curves $a\ b\ c, a'\ b'\ c'$, which are only rough approximations to circles.*

Zeuner, as is well known, replaces these two curves by equal circles the centers of which lie on the same straight line through O. Thus, $\theta - \varepsilon_1$, being the crank angle between any vector Oa' and the diametral vector, he shows that the circle diagram will be given by

$$r = a_1 \cos(\theta - \varepsilon_1).$$

In this case a_1 is the diameter of each circle or half the travel

* See Appendix A.

of the valve. His "missing quantity" can readily be put in the form

$$k + a_2 \cos(2\theta - \varepsilon_2),$$

the point from which the displacements of the valve are measured being so chosen that k is small. His determination of the constants k, a_2, ε_2, are only approximate, and the remainder of the series he, very properly for his purpose, neglected.

4. INERTIA STRESS FROM DIAGRAMS. IMPORTANCE OF HARMONICS.

While the circle diagram alone could not be considered a very good approximation to the valve motion in Fig. 1, combined with the missing quantity it would be ample in determining steam distribution. But in determining stresses the terms $a_3 \cos(3\theta - \varepsilon_3)$ . . . . would have a very sensible effect. The consideration of the stresses induced by the motion of the valve, Zeuner does not enter on. At the time his work was published there was no necessity, as the flat slide valve in which frictional resistances largely preponderated was then in almost universal use.

Since then developments, both on land and sea, have called for engines of increased power and speed, the consequent gradual evolution of engine design is too well known to require even an outline here. But comparing old and new engines of the same stroke, the increase of power, gained partly by greater speed of revolution, partly by greater diameter of cylinder and increased steam pressure, requires larger ports and valves, and greater travel of valve. Comparing a double compound engine of twenty years ago with a modern triple or quadruple expansion engine of comparatively high speed, it will usually be found that the ratio of inertia stresses to friction in the L.P. flat slide valves is greater in the former. To diminish friction the piston valve came into very common use, being fitted in many cases even to the L.P. cylinder. Here the inertia stress at once becomes the predominant one.

But further important effects result from growth of the stroke of the valve relatively to the stroke of the engine. The importance of Zeuner's "missing quantity," relatively to those parts of

the motion he retains, increases in proportion to the stroke of valve, at least approximately; and the approximateness of its expression rapidily diminishes, so that the terms after the third on the right-hand side of the equation (1) will begin to have a sensible value. Zeuner's expression for the "missing quantity" may be somewhat loosely described as the minimum value of the departure of the true valve motion from the circle diagram.

Side by side with the development in power and speed has come the very frequent adoption of non-eccentric or single eccentric valve gears already mentioned. In these the conditions for the best setting are more difficult to ascertain than in the case of the Stephenson link, and are not so well understood. Links or radius rods swinging through a large angle form parts of the gear; and, either from necessity or convenience, their length is frequently much restricted. From these several causes combined the valve movements produced by those gears often present much larger deviations from the simple harmonic motion of the Zeuner diagram than need occur with the Stephenson link. The disturbance of the inertia stresses will consequently, as we shall see presently, be greatly exaggerated. Fig. 1 offers an excellent example of these deviations.

If t is the time, ω the angular velocity of the crank in radians per second—supposed constant—and M the mass of the valve, its inertia in gravitation units is given by

$$\frac{M}{g}\frac{d^2r}{dt^2} = \frac{M\omega^2}{g}\frac{d^2r}{d\theta^2}, \quad \ldots\ldots\ldots (2)$$

since

$$\omega = \frac{d\theta}{dt};$$

or, from equation (1).

$$\text{Inertia stress} = -\frac{M\omega^2}{g}\{a_1 cos(\theta - \varepsilon_1) + 2^2 a_2 cos(2\theta - \varepsilon_2) + 3^2 a_3 cos(3\theta - \varepsilon_3) + \ldots\}.^* \quad \ldots (3)$$

*As it is more convenient, the sign of the inertia stress is changed in the sequel from that given by equation (3). Then, for a simple harmonic motion $r = a\ cos\ \theta$, the inertia will be reckoned positive when $\theta = 0$. Also, any other force acting on M will be reckoned positive when it will strain the gear in the same way as positive inertia.

Fig. 1₁.

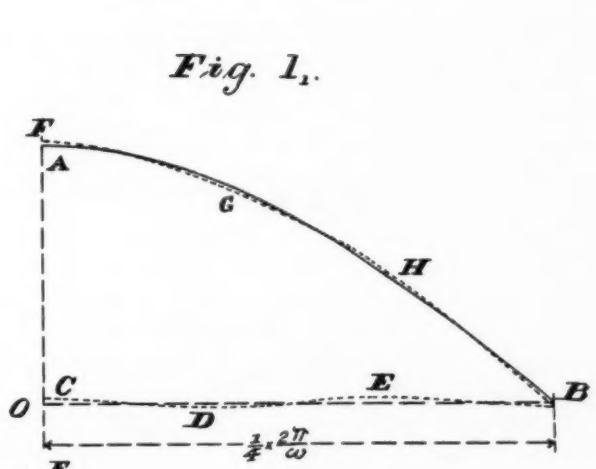


Fig. 1₂.

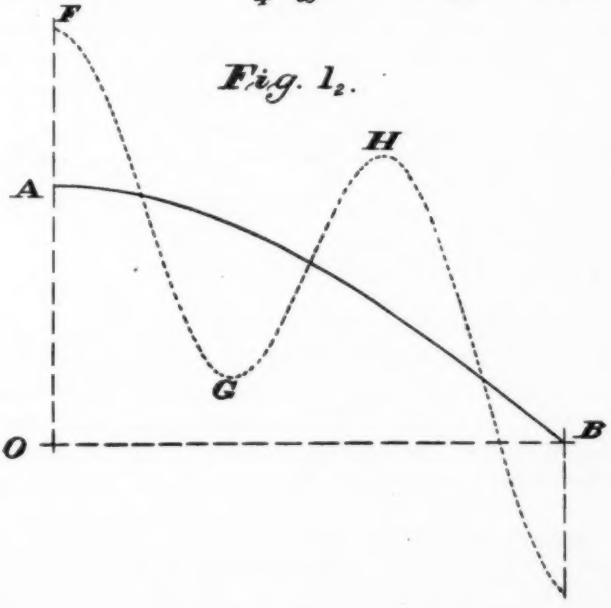
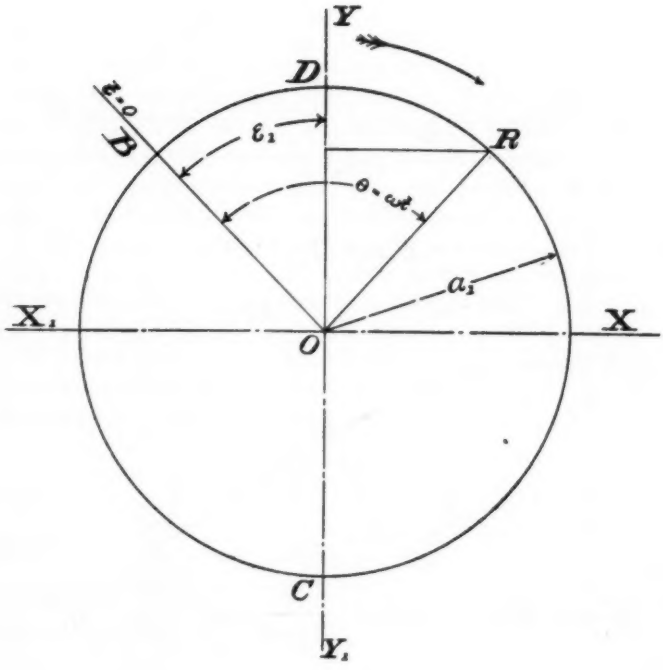


Fig. 1₃.



While the ratios of the maximum displacements, or amplitudes, due to the 3d, 4th, &c., terms of equation (1), to that of the 2d term, are

$$\frac{a_2}{a_1}, \frac{a_3}{a_1}, \text{ \&c.,}$$

the ratios of the corresponding stresses are

$$\frac{2^2 a_2}{a_1}, \frac{3^2 a_3}{a_1}, \text{ \&c.}$$

Thus, though these terms may be unimportant when determining steam distribution they may still be well worth considering when estimating stresses. In my former paper* referred to above I give a forcible illustration of this which I will repeat here.

In Fig. 1_1, OB is a time base, and the full line curve AB is a curve of cosines, $cos\omega t$. The dotted line CDE is also a curve of cosines of one-sixth the period and one-sixtieth of the amplitude of AB. If we superpose these two curves we get the dotted line FGH of which

$$r = cos\omega t + \frac{1}{60} cos6\omega t$$

is the equation.

The accelerations are given by

$$-\frac{d^2 r}{dt^2} = \omega^2 \left(cos\omega t + \frac{6^2}{60} cos6\omega t \right).$$

Thus, the curves AB and FGH, Fig. 1_2, represent to the proper scales the accelerations and forces of the movements of the same denomination in Fig. 1_1.

5. SIMPLE HARMONIC MOTION.

As we will have to deal constantly with simple harmonic motion I will, for reference, give here the well known definitions applying to it.

In Fig. 1_3 the radius OR revolves uniformly at ω radians per second, in the direction of the arrow.

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* On the Inertia Stresses of a Slide Valve. JOURNAL OF THE AMERICAN SOCIETY OF NAVAL ENGINEERS, Vol. VI, p. 272.

From R drop a perpendicular AR on YY_1. Then as OR revolves, A will vibrate equally on each side of the center O between the points C and D.

The motion of A is simple harmonic.

Let OR coincide with the radius OB when $t = 0$.

Angle $BOD = \varepsilon_1$, $BOR = \theta$.

$OR = a_1$.

Then $OA = r = a_1 cos\,(\theta - \varepsilon_1) = a_1 cos\,(\omega t - \varepsilon_1)$,

as $\theta = \omega t$.

If T_1 is the time of a complete revolution of OR and, consequently, of a complete vibration of A,

$$\omega T_1 = 2\pi \text{ or } T_1 = \frac{2\pi}{\omega}.$$

Hence $$r = a_1 cos\left(\frac{2\pi}{T_1} t - \varepsilon_1\right).$$

The *Amplitude* of the simple harmonic motion is $a_1 = OD$.

The *Period* is T_1.

The *Epoch* is ε_1.

The *Phase* is the ratio $\frac{\text{Angle } X_1OR}{2\pi}$; or "the fraction of the whole period which has elapsed since the moving point last passed through its middle point in the positive direction."

What I have defined here as *Epoch* is more properly the *Epoch in Angular Measure.* The *Epoch* as usually defined is the time of describing the angle BOD; but as the angle itself only is referred to in the sequel, the shorter name is used.

The third term of equation (1)

$$a_2 cos\,(2\theta - \varepsilon_2)$$

is a simple harmonic motion of amplitude a_2 and angular velocity 2ω. Its periodic time is

$$T_2 = \frac{2\pi}{2\omega} = \frac{T_1}{2},$$

performing two complete cycles for one of the term

$$a_1 cos\,(\theta - \varepsilon_1).$$

The epoch of the third term is ε_2.

Similar remarks apply to all the succeeding terms.

The vibration $a_1 cos(\theta - \varepsilon_1)$ I have called the vibration of the 1st period; $a_2 cos(2\theta - \varepsilon_2)$, the vibration of the 2d period; $a_n cos(n\theta - \varepsilon_n)$, the vibration of the nth period.

6. RIGID AND ELASTIC GEAR.

As the elasticity of the gear is in no way involved in a valve motion diagram like Fig. 1, the resulting stress deduced from it is the same as we would have if each part of the gear were perfectly rigid, but the joints perfectly mobile without friction or slack. This is what is meant when "*rigid gear*" is written. Thus $a_1\ a_2\ a_3$, ... are the *rigid gear* amplitudes of the 1st, 2d, 3d, ... periods, respectively. Equation (1) gives the *rigid gear* valve motion; equation (2), the *rigid gear* inertia stress, and so on.

In contradistinction the *actual* (or *elastic gear*) amplitudes of the various periods, *actual* valve motion, *actual* stress, and so on are written.

7. COURSE OF INVESTIGATION.

The deduction of the stresses from the valve ellipse, given in my previous paper, not being suitable for the present purpose, a method of deducing the rigid gear stresses from the polar diagram is first given, in Chapter II. This is a necessary preliminary to treating the effects of elasticity, but is also important in itself as showing how greatly the stresses due to simple harmonic motion may be exceeded however rigid the gear. In the full ahead gear of the *Newark*, Fig. 1, the stress at a' is shown to rise about 86 per cent. above that due to a simple harmonic motion which has the same total travel as the valve.

In the same chapter I give, as an illustration, the results of the harmonic analysis of two kinds of motion—that is, finding the equation of motion in the same form as equation (1). One of these is applied in Chapter X when discussing the theory of the Marshall valve gear.

If the valve is submitted to no frictional or other resistance, and if we may suppose the gear equally elastic in all configurations as the crank revolves, the actual motion that would result

if the rigid gear motion is expressed by equation (1), is given by a well known solution of the differential equation of motion. The amplitudes of the different period rigid gear motions, when non-synchronous, are all changed in definite proportions depending on the ratio of each particular period to the natural period of the gear. The epochs are not changed.

Where there is synchronism—the period of any one of the terms of equation (1) coinciding with the natural period—the transfer of energy from the gear to the valve produces a vibration of natural period and of continually increasing amplitude. This transfer of energy is effected through a change in the epoch of the actual synchronizing vibration from that of the rigid gear vibration. Thus, no gear could stand, as we may be sure synchronism will be closely enough established over considerable periods at some time during the running of the machine.

Hence, in order that any gear may run successfully, there must be some agent absorbing this continual supply of energy and keeping the induced stress within practicable limits. The most important is the action of friction; and to the effect of friction, constant in amount, opposing the motion, the investigation is almost entirely devoted. For more completeness, the known solution for friction varying in proportion to the velocity is given in Chapter XIII.

The viscosity of the metal of the gear is a natural action of the utmost importance in mechanics; an importance which, I fear, is far from generally or sufficiently recognized. This is the molecular action by which the energy of any vibrating mass is gradually transformed; so that, even when there is no external resistance, the vibrations will soon subside. Thus, the inaudible vibrations of a tuning fork excited in a vacuum, while continuing a little longer than in air, soon disappear. The law of this transformation has been carefully studied by several experimenters and, while it is a complicated phenomenon, this law may, with sufficient accuracy for the present purpose, be stated very simply thus, *a definite and constant fraction of the energy of vibration is transformed during each vibration.*

The value of this fraction used in the deductions in Chapter

XIII is .004, or rather under half a per cent. The experiments referred to in Chapter IV, were all on the torsion of wire. Experiments have been made to determine this rate of absorption of energy for other kinds of strain, by the late Prof. Bauschinger of Munich, but I have been unable to obtain his results. The law of subsidence of vibrations seems to be the same for other kinds of strain as for torsional, hence the nature of the results in Chapter XIII will be unaffected, though the coefficient .004 may be considerably in error. The effect of a large change in this value is exhibited.

Returning to the problem of the elastic gear with constant frictional resistance opposing the motion of the valve, after some general considerations I treat, in Chapter V, the case of the flat slide valve with friction greatly in excess of the rigid gear inertia. The combined action of friction, inertia, and elasticity is shown to have an important effect in increasing the inertia and thus raising the total stress. Examples of this are given in Table V.

Before introducing the general theory of the elastic gear motion with constant friction I give, in Chapter VI, three simple illustrations of frictionally-resisted motion. These bring out several important features, especially regarding the transfer of energy to or from the actual synchronizing vibration of the valve.

The *3d Illustration* is of special interest. In it the only two rigid gear vibrations supposed present are those of 1st and natural period, and the latter performs an odd number of complete vibrations for one of the former. The epochs are such that when the 1st period rigid gear vibration is at its maximum positive displacement the *rigid gear* synchronizing vibration is passing through zero displacement in the positive direction. The *actual* synchronizing vibration, it is shown, will then be at its maximum negative displacement. It is proved that until the synchronizing vibration has grown till its maximum inertia stress more than equals that of the 1st period vibration, both the friction and the gear transmit energy to the valve. Thus the actual synchronizing vibration is increased from both sources. That the conditions here are stable is shown by a proposition in Chapter VII. As nearly the same conditions may frequently

arise in practice we have an important cause of increased stress. To produce this result it will be shown that it is not even necessary that there should be a term of natural period in the rigid gear motion.

I then, in Chapter VII, treat the motion more generally, but any very concise statement of results could hardly be made satisfactory. The principal questions requiring consideration are: 1st. The transfer of energy to or from the actual synchronizing vibration of the valve, both by means of friction and the action of the gear; 2d. The changes of epoch and amplitude of the actual synchronizing vibration which, due to friction, take place suddenly at each reversal of the valve and, due to the action of the gear, are progressive between the reversals. The total resulting changes of epoch and amplitude from these two causes must be zero over each revolution of the engine, if the same events are to be repeated exactly during successive revolutions.

Restriction is in Chapter VIII made to the really important case—in which the ratio of the friction to the rigid gear inertia at reversal is small.

The transfer of energy and change of epoch per revolution, through friction, are there given in two sets of curves, from which those referring to any special valve motion can readily be deduced, as shown in Chapter IX.

In Chapter X is given an analysis of the motion of a valve actuated by the Marshall gear, showing how the amplitude and epoch of the short period rigid gear terms may be readily estimated.

Chapter XI deals with the application of the foregoing theory to the L.P. valve gear of U. S. S. *Newark*. In the first two official trials this gear broke down, while in the third, being slightly linked up, it was entirely successful.

We have first to calculate the natural period of the valve, depending on its mass and the rigidity of the gear; then the amplitude and epoch of the synchronizing vibration; next, the probable amount of friction of the valve.

The consequent intensity to which the actual synchronizing

vibration attained is readily determined, which, added to friction and the inertia of the non-synchronizing harmonic elements, gives the total stress. The whole course of the calculation is clearly set down, along with some other quantities, in Table X.

The calculations show that the amplitude of the actual synchronizing vibration decreases very rapidly as the angle of the reversing lever is diminished; that is, as the gear is linked up. This is entirely in accordance with experience. Indeed, so rapid is the change that, though the friction of the valve can only be roughly estimated, the safe limit of angle of the reversing lever can be very sharply determined.

At the longest cut off, .76 of the stroke, Table X shows that the safe limit has been very decidedly passed; the friction required to keep the inertia stress of the actual synchronizing vibration down to the value there given is far beyond its probable value. Thus the total stress on the part which broke would actually rise far above that given for this cut off, and even that is dangerously high.

At .725 cut off the value of the friction given in the table is probable, and the resulting total stress on the part which broke is somewhere near the danger line.

At .65 cut off, with a probable value of the friction, the total stress seems a perfectly safe one.

The results of the trials as given in the official reports entirely agree with these results of calculation. The gear seems to have broken down at once at .76 cut off; at .725 cut off the trial lasted for several hours at full speed before the gear broke; at .65 cut off the full power trial was quite successful.

The sharpness of definitions of the safe angle of reversing lever is shown by the change from .76 to .65 cut off, making a consequent change in the resulting stresses from far beyond the limit of safety to well within it, while the actual angular change is only 6° 19′.

The growth of amplitude of the synchronizing vibration is remarkable, that of the actual vibration at .65 cut off being so great as 32 times the amplitude of the corresponding rigid gear vibration, when the friction has a probable value.

The gear of the new engines of U. S. S. *Chicago* is treated in Chapter XII. In the light of the present investigation an apparently slight alteration was made in the gear, which allowed the ahead angle of the reversing lever to be considerably diminished. The improvement in the smoothness of the valve curves over those of the *Newark* is very marked, and the report of the working of the gear is eminently satisfactory.

In the same chapter, after a few remarks on various gears, the somewhat important question of the effect of stiffening any gear is discussed.

Chapter XIII opens with the solution of the valve motion resulting from an elastic gear when the friction of the valve is a proportional to the velocity of sliding.

After this the effect of viscosity acting according to the law stated above and with the coefficient .004, is examined. It is shown that the influence which this agent would have had, if taken into account in the preceding calculations regarding the *Newark*, is entirely neglible.

The motion of the valve, supposing only viscosity present, is treated very briefly. This solution brings out very clearly the limiting want of smoothness—that is, the magnitude of the short period harmonics—for a given stiffness of gear, which is admissible if the inertia stresses are to be kept within practicable limits. But, perhaps, the greatest interest of this solution arises from the light it throws on those actions where there is no sliding of surfaces in contact, and, consequently no absorption of energy by friction. We may choose from a multitude of examples the three following:

The structure of a building in which machinery is at work.

The framing of the machine itself, and

The delicate carbon filament of an incandescent electric lamp standing near a machine in motion, or otherwise submitted to synchronous vibration.

But for viscosity all these would be disintegrated.

In Japan, where the houses are frequently subjected to earthquake shocks of considerable energy, a layer of shot is often introduced between the house and the foundation. The function

of this shot is, no doubt, to prevent the full effect of the shock being transmitted to the structure above it. But, as the relative motion of the house and its foundation must be exceedingly small, any result can only be due to the modifying of the effect which would otherwise be produced by the vibrations of small amplitude; and it seems to me most probable that one essential factor in the success of the device is the introduction of a means of absorbing the energy of, and thus damping out, these small vibrations which, by synchronism with one or other of the natural periods of the upper structure, might produce disastrous effects.

I have not treated the effects of fluid resistances in absorbing energy. The effect of air or other gas in which the machine or structure is moving, there being no powerful check on the flow of the gas, must usually be quite negligible. If water or other liquid is the agent the effect may be great. For instance, the water in which a ship floats may be a very important factor in quelling the vibration set up by the engine; but what the relative importance of this fluid resistance and the viscosity of the ship and its cargo in absorbing energy is, has never been determined. One or other is essential to the preservation of the structure of the ship.

In this connection we may note the use of dash-pots, in which gas or liquid, the movement of which is greatly impeded, is used to absorb the energy of a suddenly released gear—energy which would otherwise be disastrous by producing shock or setting up severe vibration.

The final chapter is, to some extent, a return to the larger problem stated in §1. The various causes of increased stress may act in the case of any elastic structure or gear, and I seek to show that it is of very common occurence. This appears to me the principal cause of factors of safety for live loads being rarely below 8 or 10 while in Wohlers' experiments, in which inertia effects were carefully avoided, the necessary factor never rises above 3.50 for a repeated stress which varies between two maxima, equal but opposite. For a smaller range of stress the neccessary factor is lower. Though there are reasons—such as

the difficulty of making exhaustive calculation even of the stresses usually taken, redunancy of parts, and so on—for increasing the factor of safety, these are, after all, minor causes and leave a great discrepancy still to be accounted for. Even with the large apparent factors we have breakages which cannot be traced to unsound material. The existence of conditions which at least double the rigid gear stress cannot be rare; and in exceptional cases the stress will be much more than doubled. This seems adequate to fill the gap between Wohler's experiments and practice.

In conclusion, it is shown how, without repeating laborious calculations, the results obtained may be utilized to avoid unsatisfactory conditions and consequent high stress. The examination of other troublesome gears, if properly tabulated, would no doubt gradually supply useful data.

CHAPTER II.

DETERMINATION OF INERTIA STRESSES FROM EXACT POLAR DIAGRAM—RIGID GEAR. HARMONIC ANALYSIS.

8. SOLUTION OF PROBLEM.

In Fig. 1, OX represents the position of the crank when turned toward the cylinder, and the displacement of the valve is then Oa. When the crank has turned through an angle XOb, the corresponding displacement of the valve is Ob, and so on.*

Let us estimate the stress at any point b. $Ob = r$. What is required is to determine the value of $d^2r/d\theta^2$. Take bd normal to the curve at b, d being the center of curvature. From d draw de perpendicular to Ob. Let $bd = \rho$, $de = p$, and the angle $Obd = \varphi$. The following relations will be found in most works on the Differential Calculus,—

$$\left.\begin{aligned} \tan\varphi &= \frac{1}{r}\frac{dr}{d\theta} \\ \rho &= \frac{\left\{r^2 + \left(\frac{dr}{d\theta}\right)^2\right\}^{\frac{3}{2}}}{r^2 + 2\left(\frac{dr}{d\theta}\right)^2 - r\frac{d^2r}{d\theta^2}} \end{aligned}\right\} \quad \ldots\ldots \quad (4)$$

* See Appendix A.

where ρ is positive when the center of curvature and the origin O are on the same side of the tangent at the point (r, θ). From equations (4) we deduce

$$\rho = \frac{r\left\{1 + \frac{1}{r^2}\left(\frac{dr}{d\theta}\right)^2\right\}^{\frac{3}{2}}}{1 + \frac{2}{r^2}\left(\frac{dr}{d\theta}\right)^2 - \frac{1}{r}\frac{d^2r}{d\theta^2}} = \frac{r\left\{1 + tan^2\varphi\right\}^{\frac{3}{2}}}{1 + 2tan^2\varphi - \frac{1}{r}\frac{d^2r}{d\theta^2}}$$

$$= \frac{rsec^3\varphi}{2sec^2\varphi - 1 - \frac{1}{r}\frac{d^2r}{d\theta^2}}.$$

Thus
$$-\frac{1}{r}\frac{d^2r}{d\theta^2} = \frac{r}{\rho}sec^3\varphi - 2sec^2\varphi + 1, \quad . \quad . \quad . \quad . \quad (5)$$

This determines $d^2r/d\theta^2$, and equation (2) gives the inertia stress. If O is near the center of motion, the right-hand side of equation (5) will usually be positive ($d^2r/d\theta^2$ negative), as this indicates that the stress of the gear is urging the valve towards O.

It will be useful to note that when $\varphi = 0$ equation (5) takes the simple form:

$$-\frac{1}{r}\frac{d^2r}{d\theta^2} = \frac{r}{\rho} - 1, \quad . \quad . \quad . \quad . \quad . \quad . \quad (6)$$

To estimate approximately the degree of accuracy with which ρ could be measured, the following two curves were described, the radius vectors being determined graphically for every 10°,

$$r = 4cos\theta + .17cos2\theta,$$
$$r = 4cos\theta + .17cos2\theta + .1cos3\theta,$$

the unit of length being 1 inch. The radii of curvature were then measured from the curves and afterward calculated. The result seems to indicate that with care it should be possible to estimate ρ within 5 per cent. from the comparatively small diagram used. The degree of accuracy attainable will vary with the regularity of the curve.

Sec φ, when small, will be most readily found by measuring p. We then have,—

$$sec^2\varphi = \frac{\rho^2}{\rho^2 - p^2} = \frac{\rho^2}{(\rho - p)(\rho + p)}. \quad . \quad . \quad . \quad . \quad (7)$$

Using the latter form allows both $sec^2\varphi$ and $sec^3\varphi$ to be very readily calculated logarithmically.

The estimation of $sec\varphi$ is liable to two sources of error, and as it is raised to the second and third powers in the formula a small error in its determination will almost invariably produce a much larger error in the final result.

1st. There may be an error in the determination of the direction of the normal bd.

2d. There may be an error in measuring the lengths from which $sec\varphi$ is determined, whether the method indicated in equation (7), or some other, is used.

9. MODIFICATIONS OF METHOD.

These sources of error may be eliminated by the following simple graphical process which can readily be performed with great accuracy:—

The inertia stress of the valve depends only on its acceleration or time rate of change of velocity. Hence, if another equal mass performed movements which only differed from those of the valve by a constant velocity in the line of motion, the inertia stresses would be the same. We may thus add any constant value of dr/dt or $\omega dr/d\theta$ to the valve motion. A value of $dr/d\theta$ which, added to the displacements of the valve in Fig. 1, will make φ very small at any point can readily be determined,—

Suppose it is desired to determine the stress at a' and to change the curve so that φ will become very small there. Draw through a' the circle $fghk$ with center O, the angular intervals,—which we may call $\delta\theta$—between the consecutive radius vectors Of, Og, Oa', Oh, Ok, being equal. The corresponding radius vectors to the curve $a'\ b'\ c'$ . . . are Of', Og', Oa', Oh', Ok'. Take $\frac{1}{2}(gg' + hh')$ as δr.

If then we increase Og' by δr, Of' by $2\delta r$, &c.; and decrease Oh' by δr, Ok' by $2\delta r$, &c.; we will get a new curve $f_1 g_1 a'\ h_1 k_1$ . . . which differs from $a'\ b'\ c'$ . . . by a constant value of $dr/d\theta$ but for which φ is very small at a'. If through more than usual irregularity of the curve this change should leave φ greater than desired, any further required change will be readily found in the same way from the new curve. The radius of

curvature will be changed in such a way as to leave the stress the same, and equation (6) will now be used.

In all cases which we need consider, the value of ρ for the transformed curve $f_1g_1a'h_1k_1$ . . . . will be smaller than that for the original curve $f'g'a'h'k'$ . . . . It may be found to make sure of the accuracy of the measurements if we again change the curve by adding a constant quantity to each of the radius vectors of $f_1g_1a'h_1k_1$ . . . . , thus obtaining a curve $f_2g_2a_2h_2k_2$ . . . . The latter of these two curves will have the larger radius of curvature. We are obviously at liberty to make this constant addition to the radius vectors, as the change merely alters the arbitrary point from which displacements are measured.*

There is another transformation of the valve motion curve which seems capable of giving very accurate results and which requires no measurement of the radius of curvature. If in equation (6) we put ρ infinite, we get,

$$\frac{d^2r}{d\theta^2} = r. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (8)$$

Thus, after bringing φ to zero, we must find by trial the constant change required to be made in all the radius vectors near that of the point which it is desired to measure the stress, which will change the valve curve into a straight line. This is shown in Fig. 2. Suppose *abcde* a portion of a valve curve in which φ has been reduced to zero at *c*. Draw the radius vectors *Oa*, *Ob*, &c., continuing them on the other side of the pole from the curve *abcde*. At short intervals on the far side of the pole from *c* draw lines *pq* perpendicular to *Oc*. As a first trial we may change all the radius vectors by the length cc_1 thus producing the curve $a_1b_1c_1d_1e_1$. This is seen to be convex to the pole, and consequently the shift required to bring the new curve to a straight line will be found to be greater than cc_1. We ultimately find that the change cc' gives a curve which for a considerable distance on each side of c' sensibly coincides with the perpendicular line *pq* through c'. The value of $d^2r/d\theta^2$ at *c* will then be

* See appendix B.

given by Oc'. If a larger change than cc' had been made, a curve concave to O would have resulted. By noting the change which will give a curve just sensibly convex to O and that which will give one just sensibly concave, the limits of error of the deduced stress may be approximately determined.

10. NUMERICAL EXAMPLES.

The following numerical examples are from the L.P. valve motion of U. S. S. *Newark:*

Weight of two L.P. valves and stems, . $= 2232$ pounds.

Revolutions of engine, ... $= 127$ per minute.

$$\omega = \frac{2\pi \times 127}{60} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad = 13.30 \text{ radians per sec.}$$

$$\frac{M\omega^2}{g} = \frac{2232 \times 13.3^2}{32.2} \quad . \quad . \quad . \quad . \quad . \quad . \quad = 12261.$$

Example I.

Stress at b, Fig. 1.

r, ... $= 4.17/12$ feet.

ρ, ... $= 2.16/12$ feet.

p, ... $= .65/12$ feet.

$Sec^2\varphi$, from equation (7) $= \dfrac{2.16^2}{2.16 - .65} \quad \dfrac{2.16^2}{2.81 \times 1.51}$

Log 2.16	$= .334454$	*Log* 2.81	$= .448706$
	2	" 1.51	$= .178977$
" 2.16^2	$= .668908$	" 2.81×1.51	$= .627683$
" 2.81×1.51	$= .627683$		
" $Sec^2\varphi$	$= .041225$	$Sec^2\varphi$	$= 1.0996$
" $Sec\varphi$	$= .020612$		
" $Sec^3\varphi$	$= .061837$	$Sec^3\varphi$	$= 1.1530$

$-\dfrac{d^2r}{d\theta^2}$ in feet, from equation (5)

$$= \frac{4.17}{12}\left\{\frac{4.17}{2.16} \times 1.1530 - 2.1992 + 1\right\}$$

$$= \frac{4.17 \times 1.0267}{12} = .357.$$

$$\text{Stress} = -\frac{M\omega^2}{g} \cdot \frac{d^2r}{d\theta^2} = 12261 \times .357 = 4377 \text{ pounds.}$$

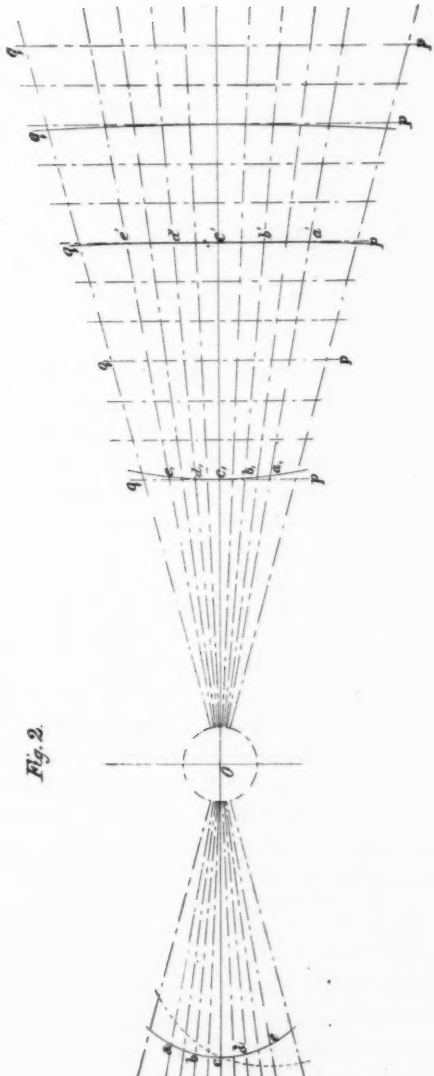
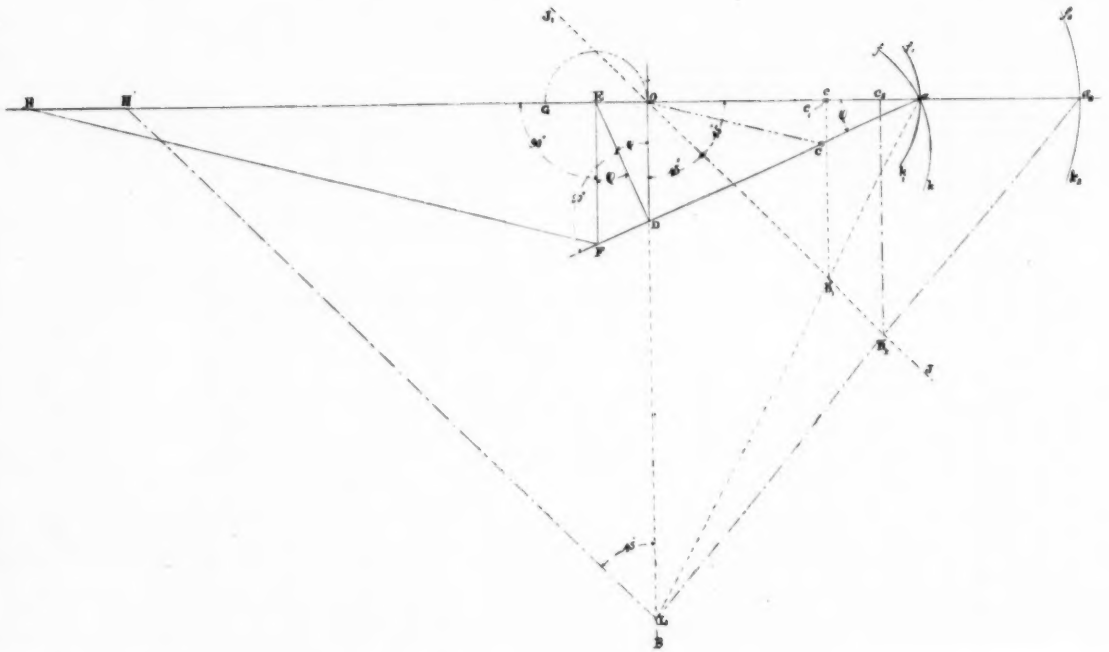


Fig. 2.

Fig. 3.



Example II.

	From curve $f'g'a'h'k'$	From curve $f_1g_1a'h_1k_1$	From curve $f_2g_2a_2h_2k_2$
Stress at a'			
r in feet	4.18/12	4.18/12	6.60/12
ρ in feet	1.68/12	1.45/12	2.94/12
p in feet	.68/12	0	0
$sec^2\varphi$	1.1959	1	1
$sec^3\varphi$	1.3079	1	1
$-\frac{d^2r}{d\Theta^2}$ in feet	$\frac{4.18}{12}\left\{\frac{4.18}{1.68}1.3079-2.3918+1\right\}$ $=.649$	$\frac{4.18}{12}\left\{\frac{4.18}{1.45}-1\right\}$ $=.656$	$\frac{6.60}{12}\left\{\frac{6.60}{2.94}-1\right\}$ $=.685$
Stress in lbs.	$12261\times.649$ $=7957.$	$12261\times.656$ $=8043.$	$12261\times.685$ $=8399.$

The stress has been calculated at a' as it is a maximum there. The fact that this maximum does not occur at the greatest displacement shows that the motion is not simple harmonic. As the pole O has been chosen in Fig. 1 very near the middle of the valve travel, the disturbances of the simple harmonic motion are also shown by the departure of the valve curves from circles.

In *Example II* the stresses at a', calculated from the different curves, differ, the greatest being almost $5\frac{1}{2}$ per cent. greater than the smallest, but this agreement is very close as the final result is so largely affected by a slight change in the estimate of the radius of curvature, and to make this estimation approximately correct requires the greatest care. When the position of a maximum has been approximately discovered the points of the valve motion curve should be determined for every two or three degrees of crank angle for a considerable distance on each side of the maximum. A light sharp line drawn with a sweep through these points will serve to discover and correct slight errors in their position, and will greatly facilitate the correct determination of the center of curvature. At any maximum point it is well to determine the stress in the three ways given in *Example II* and also as in Fig. 2. The first determinations will usually differ by considerable amounts, but having several values will guide the further examination of the curves to discover a truer value of the stress. Immediately on each side of a point of maximum curvature the circle of curvature, tangent at that point, lies wholly within the curve. Hence, if the deter-

mined points are wide apart there is a danger that ρ will be over estimated, as the tendency is to make the circle pass through three consecutive points at least.

The travel of valve in *Examples I and II* is 8.85 inches. Had the motion been simple harmonic the maximum stress would have been

$$\frac{M\omega^2}{g} \cdot \frac{8.85}{2 \times 12} = \frac{12261 \times 8.85}{24} = 4521 \text{ pounds.}$$

The maximum estimate in *Example II* exceeds this in the proportion $\frac{8399}{4521} = 1.86$. That is, the small disturbances of the simple harmonic motion nearly double the stress.

If, at any point, we suppose the circle of curvature to represent the valve motion, the stress at this point will obviously remain unchanged. Should the circle pass through the pole O it would represent a simple harmonic motion and the stress could then be calculated directly from the displacement of the point. For instance, if the circle of curvature at b, Fig. 1, passed through O the stress would be

$$= \frac{M\omega^2}{g} \times Ob. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (9)$$

The circle at b, shown dotted, actually passes very near the pole and the stress in *Example I* is found to be only from 2 to 3 per cent. in excess of that given by equation (9). Thus we can readily judge, either from the smallness of the radius of curvature, or by the nearness of the approach of the circles of curvature to the pole, at what points the stress is excessive. Indeed, the relation of the circle and pole could be made the foundation of a modification of the above modes of estimating stress, though probably not one so advantageous as those given. For, from what has been just said, it is evident that if all the points in the neighborhood of a', for instance, were gradually and equally shifted outward, the changing circle of curvature would approach the pole and when it passed through it the corresponding radius vector would be used in equation (9) for calculating the stress.

11. MR. J. MACFARLANE GRAY'S GRAPHICAL METHOD.

The results found numerically in *Examples I* and *II* may readily be obtained graphically. This simple graphical process is due to Mr. J. Macfarlane Gray who most kindly communicated it to me as a "sympathetic response" to a short note I had written him on the subject of the present investigation.

In Fig. 3, O is the pole and fak a portion of the valve curve from which the stress at a is to be estimated. C is the center of curvature for the point a. Through O draw OB at right angles to Oa. The line aC produced cuts OB in D. Through D draw DE at right angles to aC, cutting Oa produced in E. Again through E draw EF at right angles to Oa, cutting aC in F. Through F, parallel to Oc, draw FH cutting Oa produced in H. Find the point G in Oa so that $OG = 2OE$.

$Oa = r$, and the angle $OaC = \varphi$.

Hence $OD = r\tan\varphi$, and $OE = ODtan\varphi = r\tan^2\varphi$. Thus $OG = 2r\tan^2\varphi$.

Also $aD = r\sec\varphi$; $aE = aD\sec\varphi = r\sec^2\varphi$;

and $aF = aE\sec\varphi = r\sec^3\varphi$.

By similar triangles $\dfrac{aH}{aF} = \dfrac{Oa}{aC} = \dfrac{r}{\rho}$;

hence, $$aH = \frac{r}{\rho} \cdot r\sec^3\varphi.$$

Since, as already shown, $OG = 2r\tan^2\varphi$,

we have $$HG = aH - OG - Oa = \frac{r}{\rho} \cdot r\sec^3\varphi - 2r\tan^2\varphi - r.$$

$$= r\left(\frac{r}{\rho}\sec^3\varphi - 2\tan^2\varphi - 1\right)$$

$$= r\left(\frac{r}{\rho}\sec^3\varphi - 2\sec^2\varphi + 1\right) = -\frac{d^2r}{d\theta^2},$$

by equation (5).

The procedure must be modified when φ is zero or very small. f_1ak_1 is a portion of the valve curve in which φ has been brought nearly to zero in the way previously indicated. The center of curvature is now C_1, very near Oa. Through O draw J_1OJ making an angle of 45° with Oa; and through C_1 draw $C'K_1$ at right angles to Oa, cutting Oa at C' and J_1oJ at K_1. Join aK_1, and produce it to cut OB in L.

If φ is not much greater than, say, 10°, we may in the present case suppose $cos\varphi$ and $sec\varphi$ to equal unity; as the error—about $1\frac{1}{2}$ per cent.—is much less than the probable error in estimating φ.

This gives $aC' = aC_1 = \rho$, and reduces equation (5) to equation (6). We therefore have, in Fig. 3, $OC' = r - \rho = C'K_1$, since $aOJ = 45°$. By similar triangles,

$$\frac{OL}{Oa} = \frac{C'K_1}{C'a},$$

or
$$OL = \frac{r}{\rho}\left(r - \rho\right) = r\left(\frac{r}{\rho} - 1\right) = -\frac{d^2r}{d\theta^2},$$

by equation (6).

When φ is greater than 10° the earlier modification of the graphical process is available.

If we alter the curve f_1ak_1 by making constant additions to the radius vectors, in the way we have obtained the curve $f_2g_2a_2h_2k_2$ . . . . , Fig. 1, we may readily follow the change of position of the center of curvature. Suppose, in Fig. 3, all the radius vectors to be changed by aa_2, giving a curve $f_2a_2k_2$. For this new curve $d^2r/d\theta^2$ will be the same as for f_1ak_1, since the inertia stress is not altered. Hence if we join a_2L, cutting J_1oJ in K_2, and draw K_2C_2 perpendicular to Oa, cutting it in C_2, C_2 is the center of curvature of $f_2a_2k_2$, if φ is zero for f_1ak_1 and $f_2a_2k_2$; or a_2C_2 is the projection of the radius of curvature at a_2 if φ is small.

If through L we draw LH' parallel to J_1oJ, cutting Oa in H', we see at once that the effect of making the constant shift of radius vector aH' is to obtain a curve which has no curvature at H'—since the point in which LH' and J_1oJ intersect is infinitely removed.

As $OLH' = 45°$, $OH' = OL$, or

$$\frac{d^2r}{d\theta^2} = r$$

as in equation (8). Thus the meaning of the process in Fig. 2 is made clearer; and, referring to the letters in Fig. 2, it becomes

immediately evident that if C and C_1 lie on the same side of C' their concavities are both turned in the same direction. If they lie on opposite sides, these concavities are turned oppositely. That is, in the curves for which φ is zero, which are deduced from one another by constant changes of the radius vectors, if one is convex to the point C_1' Fig. 2, at which the curvature is zero, all are convex; and if one is concave, all are concave.

12. HARMONIC ANALYSIS OF TWO MOVEMENTS.

Another method of determining the stress consists in the application of Fourier's Theorem to resolve the valve motion curve into its simple harmonic constituents, giving a result in the form of equation (1). It can be shown that one, and only one, such resolution is always possible. In many works in Mathematics and Physics Fourier's Theorem will be found discussed and applied, and I do not propose to enter on it here. In a recent paper* I had occasion to give an example of the application of this method to the harmonic analysis of a curve.

This analysis is of the deepest theoretical interest and importance and I refer to it as equation (1) is the foundation of the present investigation. Its application to any curve, without the use of machines invented to make the resolution, involves considerable labor, and it would hardly be possible to obtain more than three, or at most four, simple harmonic elements of a valve curve with any degree of accuracy, as the amplitudes of the successive elements diminish so rapidly.

I will give here, for future reference, the results of the harmonic analysis of two simple cases of periodic motion.

Example I.

Motion of a point attached to the crosshead of an ordinary connecting rod engine.†

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* See "Engineering," Vol. LXIV, page 543.

† See paper on Crank Shafts by the present writer; JOURNAL OF THE AMERICAN SOCIETY OF NAVAL ENGINEERS, Vol. IX, p. 505, equation (71); or that on the Analysis of the Inertia Forces of the Moving Parts of an Engine, "Engineering," Vol. LXIV, p. 511, equation (3).

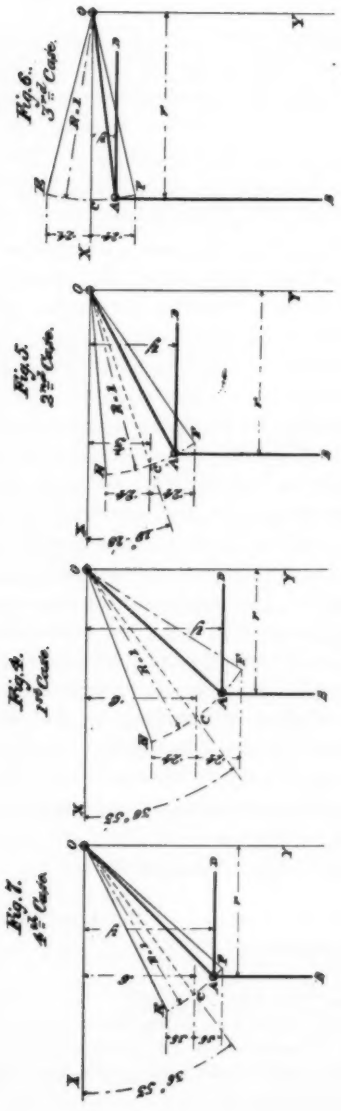
Fig. 6.
3rd Case.
Fig. 5.
3rd Case.
Fig. 4.
1st Case.
Fig. 7.
2nd Case.

r = the displacement of the point from its top stroke position in a vertical inverted engine.

R = the radius of the crank.

θ = the crank angle, measured from top center.

1st Case. Length of connecting rod 4 cranks.

2d Case. Length of connecting rod 5 cranks.

1st Case. $$r/R = 1.0632 - cos\theta - 10^{-2} \times 6.35\ cos2\theta + 10^{-4} \times 2.6\ cos4\theta - 10^{-7} \times 19\ cos6\theta. \quad . \quad . \quad (10)$$

2d Case. $$r/R = 1.0504 - cos\ \theta - 10^{-2} \times 5.05\ cos2\theta + 10^{-4} \times 1.3\ cos4\ \theta - 10^{-7} \times 6\ cos6\theta. \quad . \quad . \quad (11)$$

If the connecting rod were infinite the motion of the point would be simple harmonic, all the terms of equations (10) and (11) after $cos\theta$ disappearing.

This example is introduced principally to show that while the change in the length of the connecting rod makes no change in the amplitude of the 1st period, its influence on the higher period motions is greater the higher the period. Thus:

TABLE I.—COMPARISON OF AMPLITUDES, EQUATIONS (10) AND (11).

Period.	1st.	2d.	4th.	6th.
Ratio of amplitude, equation (11), to amplitude, equation (10).....	1	.795	.500	.316

Thus a slight diminuition in the angular swing of the connecting rod has a great influence on the motions and forces of high period. This fact will be found to be of the utmost importance as we proceed and is again illustrated in the next example.

Example II.

In Figs. 4 to 7, OA is an arm pivoted at O, the outer end A being actuated by the rod AB supposed infinite and perpendicular to OX. C is the mean height of the point A, its distance below OX being k.

$oA = R$.

$y =$ the variable displacement of A from OX.

$r =$ the variable displacement of A from OY, perpendicular to OX.

The rod AB is supposed to give the point A the motion,

$$y = k + a\cos\theta,$$

θ as before being the angle of the crank which actuates AB.

The limits of the swing of OA are given by OE and OF.

If the horizontal rod AD, infinite in length, be supposed to actuate a valve, the arrangement corresponds to a simple Marshall gear in which the rod to the excentric and that to the valve are made infinitely long. If the lead is constant, C will be the center of the reversing shaft carrying the reversing lever OC.

We have the following cases:

TABLE II.—VALUE OF k/R AND a/R, EQUATIONS (12) TO (15).

Case.	k/R.	a/R.	Angle $COX = \sin^{-1} k/R$.	Fig.
1st.	.6	.24	36°—53′	4
2d.	$\frac{1}{2}$	.24	19°—28′	5
3d.	0	.24	0°— 0′	6
4th.	.6	.15	36°—53′	7

1st Case. $$r/R = .7047 - .3546\cos\theta - .1029\cos 2\theta - 10^{-2}\times 2.81\cos 3\theta - 10^{-3}\times 8.3\cos 4\theta - 10^{-3}\times 2.49\cos 5\theta - 10^{-4}\times 7.55\cos 6\theta - 10^{-4}\times 2.31\cos 7\theta^{*} \quad . . \quad (12)$$

2d Case. $$r/R = .9175 - .1047\cos\theta - 10^{-2}\times 2.59\cos 2\theta - 10^{-3}\times 3.25\cos 3\theta - 10^{-4}\times 6.0\cos 4\theta - 10^{-5}\times 9.5\cos 5\theta - 10^{-5}\times 1.69\cos 6\theta - 10^{-6}\times 2.87\cos 7\theta \quad . . \quad (13)$$

3d Case. $$r/R = .9851 - 10^{-2}\times 1.505\cos 2\theta - 10^{-4}\times 1.67\cos 4\theta - 10^{-6}\times 2.1\cos 6\theta - 10^{-8}\times 2.7\cos 8\theta \quad . . \quad (14)$$

* See Appendix C.

4th Case. $r/R = .7680 - .1906\cos\theta - 10^{-2} \times 3.29\cos 2\theta$
$- 10^{-3} \times 5.21\cos 3\theta - 10^{-4} \times 9.0\cos 4\theta - 10^{-4} \times 1.57\cos 5\theta$
$- 10^{-5} \times 2.8\cos 6\theta - 10^{-6} \times 5.04\cos 7\theta \quad . . \quad (15)$

TABLE III.—COMPARISON OF AMPLITUDES, EQUATIONS (12) TO (15).

Line.	Period.	1st.	2d.	3d.	4th.	5th.	6th.	7th.
(*a*)	Ratio of amplitude, equation (13), to amplitude, equation (12).....	.295	.252	.116	.072	.038	.022	.012
(*b*)	Ratio of amplitude, equation (14), to amplitude, equation (12).....	0	.146	0	.0201	0	.0028	0
(*c*)	Ratio of amplitude, equation (15), to amplitude, equation (12).....	.538	.320	.185	.108	.063	.037	.022

Line (a) Table III, shows that the reduction of the angle *COX* to a little over half has the effect of reducing the amplitude and stress of the 7th period in a proportion about 25 times as great as the corresponding reduction for the 1st period.

In the 3d case the old periods do not exist, as is obvious from the symmetry of the motion in Fig. 6; but line (b) Table III, shows the same rapidly increasing ratio of reduction of amplitude for the even periods, as we pass to higher and higher periods.

Line (c) shows a similar effect produced by decreasing the angle through which the link *OA* swings. Thus, while the reduction of amplitude for the first period, per unit, length of *OA*, is nearly in the same proportion as the reduction of a/R—to a little over one-half—the amplitude of the 7th period is reduced to about one forty-fifth. This diminution of the angular swing of *OA* may obviously be obtained either by diminishing the travel of the actuating rod *AB* or increasing the length of *OA*.

I shall revert to this example in Chapter X, when treating the theory of the Marshall gear. In Table VI are given not only the coefficients of equations (12) to (15), but also those for other values of k/R and a/R.

Chapter III.

Motion of valve with elastic gear but no friction or other resistance.

13. SOLUTION OF SIMPLIFIED PROBLEM.

If we had a perfectly rigid gear to deal with, the valve motion would follow exactly that deduced from the gear diagram. But this can never be absolutely true in any actual gear, however stiff, subjected to stress; and, as we have seen that disturbances of short period may have a great effect on the stresses, the question naturally arises—What effect has the elasticity of the gear in altering inertia stresses of the various periods, and as a whole?

We shall find that in many actual cases the stresses are largely increased, since the amplitudes of some of the short period vibrations—and especially that of the synchronizing vibration—are greatly augmented in the actual motion over what they would be for a rigid gear. To estimate the total effect we must not only have regard to the elasticity of the gear, but also to friction and the viscosity of the material of the gear, both of which limit the amount of energy of vibration stored up in the valve and gear, and consequently limit the stress. But in this chapter we will consider elasticity alone, leaving out of account friction or viscosity.

The exact problem, even thus simplified, is one of great difficulty, being complicated by the varying stiffness of the gear produced by its changing configuration as the crank shaft revolves. But it is closely analogous to the simple problem of finding the motion of a weight hung to the lower end of a spring, the upper end being moved vertically in a known manner.

Suppose the elasticity of the gear or spring is such that the force required to displace the valve or weight of mass M, by a distance δr, in its line of motion, is $f\delta r$ in gravitation units. Take the mass M in pounds, f in terms of the weight of one pound, and all the lengths in feet, the intensity of gravity being g.

When M is mentioned it may be supposed that either the valve or weight suspended by the spring is referred to; as, neglecting the variation of stiffness of the valve gear, the problem of finding the motion of the mass M is the same as that of finding the motion of the valve.

If, when the engine is standing, or the upper end of the spring undisturbed, the mass M is displaced by an applied force and released, it will vibrate with a simple harmonic motion about the point from which it was displaced. As there is supposed to be no friction or other resistance, the vibration will remain of the same amplitude indefinitely. The time T of this vibration is called the natural period of the vibrating system. The angular velocity of an arm revolving in time T is given by

$$\omega_s = \frac{2\pi}{T} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (16)$$

The force $f\delta r$ producing the displacement δr is, by the well-known law for simple harmonic motion,

$$f\delta r = \frac{M\omega_s^2 \delta r}{g},$$

or

$$\omega_s^2 = \frac{fg}{M} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (17)$$

Hence if f is estimated from the drawings of the gear or by trial on the actual engine, we can at once calculate T for the known value of M.

Suppose, as before, that the valve motion which is given by the gear diagram and would be produced by a gear composed of perfectly rigid parts is expressed by equation (1); or, since

$$\theta = \omega t,$$

by
$$\left.\begin{aligned} r = k + a_1 \cos(\omega t - \varepsilon_1) + a_2 \cos(2\omega t - \varepsilon_2) \\ + a_3 \cos(3\omega t - \varepsilon_3) + \ldots \end{aligned}\right\} \quad . \quad (18)$$

or, for brevity,
$$r = k + \Sigma_1^{\infty} a_n \cos(n\omega t - \varepsilon_n)$$

The periodic time, T_1, for the term $a_1 \cos(\omega t - \varepsilon_1)$, which is also the time of revolution of the engine, is given by

$$\frac{2\pi}{T_1} = \omega. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (19)$$

The periodic times for the succeeding terms are, in order, $T_1/2$, $T_1/3$, ... T_1/n . . . . For convenience we may write,

$$T_2 = \frac{T_1}{2},\ T_3 = \frac{T_1}{3},\ \ldots\ldots,\ T_n = \frac{T_1}{n}. \quad . \quad . \quad . \quad (20)$$

From equation (19) we have,

$$\frac{2\pi}{T_2}=2\omega,\ \frac{2\pi}{T_3}=3\omega,\ \ldots\ \frac{2\pi}{T_n}=n\omega. \quad \ldots \quad (21)$$

It can readily be shown that the actual motion of the valve is given by

$$\left.\begin{aligned} r_1=k+&\frac{a_1}{1-T^2/T_1^2}\cos(\omega t-\varepsilon_1)+\frac{a_2}{1-T^2/T_2^2}\cos(2\omega t-\varepsilon_2)\\ &+\frac{a_3}{1-T^2/T_3^2}\cos(3\omega t-\varepsilon_3)+\ \ldots\ +A\cos(\omega_s t-\lambda),\\ \text{or}\quad r_1=k+&\Sigma_1^{\infty}\frac{a_n}{1-T^2/T_n^2}\cos(n\omega t-\varepsilon_n)+A\cos(\omega_s t-\lambda). \end{aligned}\right\} (22)$$

(See deduction of equation 8*d*, Appendix D.)

14. REMARKS ON SOLUTION.

Comparison of equations (18) and (22) shows that,

1st. The same set of simple harmonic motions appear in the valve motion whether the gear is practically rigid or sensibly elastic.

2d. The amplitude of each simple harmonic motion given by equation (18) appears in equation (22), altered in a definite proportion depending on the ratio T/T_n; T_n being the period of any one of the simple harmonic constituents.

3d. Each simple harmonic motion in equation (22) comes to its maximum displacement at the same time as in equation (18) since the epoch is unchanged.

4th. So long as the ratios T/T_1, T/T_2, T/T_3, ... , T/T_n, ... are small—that is, in those terms which have a period many times longer than the natural period of vibration of the valve—the components of the valve motion and the stresses due to these terms are but slightly altered by the elasticity of the gear.

5th. So long as T_1, T_2, T_3, ... T_n, ... are greater than T, the terms in equation (22) are of the same sign as the corresponding ones in equation (18)—that is $a_n/(1-T^2/T_n^2)$ is of the same sign as a_n. So that, not only do the corresponding displacements come to a maximum at the same time, but also in the same direction. Further, the actual displacement will then be larger than for a rigid gear as $1-T^2/T_n^2$ is less than unity.

A point of very high stress deduced from the valve diagram is obviously due to several of the terms in equation (18) combining to augment one another when the valve is in the position indicated by this point. If the periodic times of all the terms which constitute a very sensible amount of the total stress are longer than T, the increased terms in equation (22) will also combine to augment one another at this point; since, as we have just seen, the respective directions of displacement are the same as for the corresponding terms of equation (18). Hence, the maximum stress would be raised above what it would be for a rigid gear. We shall find the condition, that the terms with sensible magnitude should have a period longer than T, is an almost imperative criterion of successful running.

6th. If the engine runs at such a speed that, for any term

$$\frac{a_n}{1-T^2/T_n^2}\cos(n\omega t-\varepsilon_n),$$

T/T_n is very near unity, the amplitude $a_n/(1-T^2/T_n^2)$ will be much larger than a_n. The effect for a frictionless valve will be a sensible vibration with a period T_n, and the maximum stress will be largely increased.

15. SNYCHRONISM.

7th. Suppose $T_n=T$. There is now exact synchronism between the period of the nth term and the natural period of the valve.

In what follows we will have to deal very largely with this synchronizing term, and I will always designate the number of its period by s instead of n, to distinguish it from all the other terms. This has already been done for the angular velocity, ω_s, corresponding to the natural period T. Thus a_s is the amplitude and ε_s the epoch of the synchronizing vibration when the gear is rigid.

The above condition becomes $T_s=T$. As this always holds, we need, as a rule, only retain T in the sequel.

From equations (16) and (21), again changing n into s,

$$s\omega=\omega_s \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (22_1)$$

and the corresponding amplitude $a_s\,(1-T^2/T_s^2)$, equation (22), becomes infinite.

Evidently, when synchronism occurs, the infinite vibration given by the term

$$\frac{a_s}{1-T^2/T_s^2}\cos\,(s\omega t-\varepsilon_s)$$

in equation (22), ceases to be part of the solution of the problem, as the growth of the disturbance must be through the gradual communication of energy. It can readily be shown (see deduction of equation (15*d*)) that this term must now be replaced by

$$\left(B_o+\frac{a_s\omega_s t}{2}\right)\sin\,(\omega_s t-\varepsilon_s), \quad . \quad . \quad . \quad . \quad . \quad (23)$$

where B_o is the amplitude when $t=0$.

Comparison of this expression with the rigid gear term

$$\left.\begin{matrix} a_s\cos\,(s\omega t-\varepsilon_s) \\ a_s\cos\,(\omega_s t-\varepsilon_s) \end{matrix}\right\}, \quad . \quad . \quad . \quad . \quad . \quad . \quad (24)$$

or

the cause of the growing disturbance of the valve motion, clearly exhibits the rationale of the process. Term (24) expresses the displacement of this period for a rigid gear, and consequently the force exerted on the valve producing this disturbance will be proportional to the difference of the terms (23) and (24)—will be, in fact, $f\delta r_s$, where δr_s is that difference. If we may, for the moment, consider the amplitude $B_o+a_s\omega_s t/2$, in term (23), as constant during the short time T in which the motion (24) completes one cycle, we will have in terms (23) and (24) two simple harmonic motions. If the amplitudes a_s and $B_o+a_s\omega_s t/2$ are both positive, the motion (23) is quarter a period behind (24); for, when $\omega_s t-\varepsilon_s=0$, the latter has reached its maximum upward displacement, and the former is passing through its central position in the upward direction.

Consequently, the force augmenting this disturbance is at this moment urging the valve in the direction of its motion and augmenting its energy.

Following this force through a complete cycle, with or without the restriction that $B_o+a_s\omega_s t/2$ should be constant, it will be

found that while the valve motion of this period is sometimes accelerated and sometimes retarded, there is, on the whole, a positive increase of energy.

As shown in Appendix D (see deduction of equation (17*d*) the energy supplied in the cycle which commences at any time t_1 for which $\omega_s t_1 - \varepsilon_s = 2n\pi$, is

$$fa_s\left\{B_o + \frac{a_s}{2}(\omega_s t_1 + \pi)\right\}\pi. \quad . \quad . \quad . \quad . \quad (25)$$

Thus the growing energy of the disturbance is completely accounted for.

$$B_o + \frac{a_s}{2}(\omega_s t_1 + \pi)$$

is the mean amplitude, during the cycle beginning at t_1 of the resulting vibration; for while $sin\,(\omega_s t - \varepsilon_s)$ completes a cycle, $\omega_s t$ must increase by 2π. Hence if a_s' is the amplitude of the actual synchronizing vibration, and if the growth of amplitude in one vibration is so small that we may use the mean value of a_s', expression (25) becomes

$$\pi f a_s a_s'. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (25_1)$$

The geometrical meaning of this will be explained in § 44.

It is to be noted that part of the arbitrary term $Acos\,(\omega_s t - \lambda)$ of equation (22) may be included in the term (23) by properly choosing the arbitrary constants. For we may write

$$Acos\,(\omega_s t - \lambda) = B_o sin\,(\omega_s t - \varepsilon_s) + A_o cos\,(\omega_s t - \varepsilon_s), \quad . \quad (25_2)$$

where A_o is an arbitrary constant. Putting in succession $\omega_s t = 0$, and $\omega_s t = \frac{\pi}{2}$, we get,

$$Acos\lambda = -B_o sin\,\varepsilon_s + A_o cos\,\varepsilon_s,$$

and

$$Asin\lambda = B_o cos\,\varepsilon_s + A_o sin\,\varepsilon_s.$$

These two equations serve to determine A and λ for any arbitrarily-chosen value of A_o; B_o and ε_s being known.

Thus, whatever the amplitude and epoch of the arbitrary vibration of the natural period $Acos\,(\omega_s t - \lambda)$, when synchronism

is established the part of this vibration, $B_s sin\,(\omega_s t - \varepsilon_s)$, which lags one quarter period behind the rigid gear synchronizing term, begins to grow at a uniform rate $a_s\omega_s t/2$. The part $A_s cos\,(\omega_s t - \varepsilon_s)$, which agrees in phase with the rigid gear vibration, is unaffected.

All that has been said here with regard to the action of the synchronizing vibration should be most carefully studied, as it will be found of the utmost importance when we come to the practical problem of determining the motion of the valve resisted by a constant frictional force.

As the series, equation (1), giving the rigid gear motion of the valve, contains an infinite number of terms, synchronism must always occur at some time in the running of the engine. Hence we see that if there were absolutely no resistance to the motion of the valve, absorbing the communicated energy given by expression (25), no gear could run without breaking down.

This absorption of energy is produced partly by the friction of the valve and rubbing surfaces of the gear; and, especially in those cases where high natural period,—that is T_1/T, or ω_s/ω, large—and small frictional resistance occur together, viscosity of the metal of the gear may play an essential part in keeping the stresses within allowable limits.

The solution of the problem, with friction taken into account, will show exactly what the practical conditions to be fulfilled are that we may avoid a dangerous rise of stress in the gear.

CHAPTER IV.

FRICTION AND VISCOSITY OF METALS.

16. LAW OF FRICTION OF METALS.

The law of resistance to the motion of the valve varies greatly in different cases.

1st. The friction of iron on iron, when the surfaces are dry, moistened with water, or very greasy has a maximum value when the tangential force is just sufficient to start sliding. As the velocity increases the coefficient of friction falls.

The following formula is that proposed by M. Bochet in 1858,* the values of his constants adopted being those which give the greatest and least coefficients of friction in his first memoir for iron on iron dry, and wet with water. A similar law has been found to hold for other metals, and the coefficients of friction seem to differ but little.

$$\left.\begin{aligned}\text{Greatest coefficient of friction} &= \frac{.3}{1 + .009v}.\\ \text{Least coefficient of friction} &= \frac{.14}{1 + .021v}.\end{aligned}\right\} \quad . \quad . \quad (26)$$

v is the velocity of sliding in feet per second.

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* See *Comptes Rendus;* Vol. 46, p. 802, and Vol. 51, p. 974.

The *Comptes Rendus* gives only abstracts made by the author, M. H. Bochet, of his Memoirs on Friction.

The first Memoir deals with the reduction of the following experiments:

M. Morin's (1831–33). Speed generally 2 to 3 meters per second, rarely above 3, always less than 4.

M. J. Poirée's (1851). Friction of wagon wheels on rails. Speeds, 4 to 22 meters per second.

MM. Nap. Garella and H. Bochet's (1856). On the retardation of railway trains. Speeds not noted in abstract.

M. J. Poirée's (1856). Friction of wagons armed with curb shoes or skids. Speeds not noted in abstract.

Bochet proposes the formula

$$F = \frac{pK}{1 + cv}, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad A$$

where F is the friction, p the normal pressure, and v the velocity of sliding. K and c are coefficients depending on the circumstances of the experiment.

K lies between .3 when the rails are absolutely dry, and
.14 when the rails are wet.

c seems to vary slightly with K, but not to an important extent. It is given here as .03 for wheels sliding on rails, and .07 for skids on rails, v being in meters per second—.009 and .021 respectively when v is in feet per second.

He also proposes the formula

$$\frac{F}{p} = \gamma + \frac{K - \gamma}{1 + cv}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad B$$

Bochet's second Memoir gives the conclusions from further experiments of his own. Speeds from 0 to 25 meters per second. The experiments were made for a large variety of cases; but always with sledges sliding on railway rails, the runners being of various materials and the surfaces in different conditions—wet, dry, greasy, oiled, and rough or in different states of polish.

The least coefficient occurred when the surfaces were damp.

It is frequently supposed that the friction which will just prevent sliding—"the friction of rest"—instantly falls by a finite amount when sliding commences, but this has long since been disproved for metals, not only by Bochet's experiments but also by those of other experimenters.

In the present investigation we have almost exclusively to consider the intensity of friction when the valve is near the end of its travel and, consequently, the velocity very low. Hence, we may take the coefficient as constant and lying somewhere between .3 and .14.

He found variations of friction in, so far as he could judge, the same circumstances. Also the proportionality of friction to pressure was not quite true for high speeds and high pressures, though sufficiently true in ordinary practical conditions; his experiments were not sufficient to establish a law.

All his results, he states, can be brought under formula B with sufficient exactitude.

K and γ vary separately with the circumstances of the experiments. A list of their values given in the original Memoir is unfortunately omitted in the abstract. γ is always more or less inferior to K; it is not treated as negligible, as noted by Rankine and Thurston. As γ is the coefficient for very high speed, supposing the law to hold, we would expect it to be very sensible.

K for iron on iron very rough, exceptionally .60, but sometimes as low as .25. For imperfectly polished iron K was never greater than .4 and might have any value less than this depending on the state of the surface with regard to polish or lubrication, till for greasy rails and intense pressure the coefficient was very small. (Bochet's statement regarding the values of K is not altogether clear. The results seem to have been very irregular.)

c is found to be slightly variable, but according to a law which has quite eluded him; it may be taken as constant in practice. It is given as .3—almost certainly a misprint for .03, its larger value in the first Memoir; the smaller value .07, for skids on rails, is not referred to though, the experiments in the second memoir being with sledges, one would have expected this value to be confirmed. It is to be noted that in *Weisbach's Mechanics* c is given = .3 when quoting formula B. But, were this value true, Morin's experiments would have shown great variation with velocity, though the limits were narrow.

Bochet also experimented on the force necessary to start the motion and always found the friction the same as for very small motion except for wood and leather on wet and greasy rails, for which the mean value of the friction of rest was usually double that for an extremely slow speed.

See also *Friction and Lost Work in Machinery* by Prof. Thurston, Art. 139, for various other results of experiments; and *Machinery and Millwork* by Prof. Rankine, Art. 309.

The friction will be modified in the case of flat slide valves, especially by the fact that the normal force pressing the surfaces together changes somewhat with the relative position of the cylinder and valve ports and the varying steam pressures in the receivers and cylinder passages. This change of resistance depending on the position of the valve would be difficult to take into account even if it could be predicted with sufficient accuracy, and would probably lead to no result of practical importance.

17. FRICTION PROPORTIONAL TO VELOCITY.

2d. If the sliding takes place with only moderate normal pressure over a well lubricated surface the friction will be nearly proportional to the velocity. Occasionally this condition may be fulfilled approximately by piston valves; for instance, in the case of a vertical plug valve with no spring rings, which is guided by the spindle passing through a bush. The lubrication will usually, however, be most imperfect, and only a part of the resistance will depend on the volocity and that not necessarily on the first power of the velocity.

But the questions we are treating do not apply only to valve motions, and in many cases occurring in machines this law of resistance according to the first power of the velocity of sliding will be very closely realized. The case is discussed briefly in Chapter XIII.

18. VISCOSITY OF METALS.

3d. If there were no frictional resistance due to the sliding of surfaces there would still be absorption of energy from the viscosity of the metal of which the gear is composed. The law of the absorption of energy by viscosity in metals has formed the subject of experiment by various physicists. It is a highly complex phenomenon, depending not only on the nature of the metal, and mode of manufacture of the specimen tested, but largely on its immediate previous treatment, and the condition as to temperature, stress, &c., under which the experiment is made.

Suppose for instance, a wire be hung from a practically rigid support to which it is firmly attached, say, by soldering, so that no slipping can occur. Let it now be symmetrically loaded at

its lower end, set vibrating torsionally, and left to itself. Even if freed from all resistance of the air the amplitude of the oscillations will gradually subside, showing an absorption or, more correctly, a transformation of the energy of the vibrating system due to the internal action of the molecules of the wire. If the limits of strain of the wire on each side of the original position of equilibrium are wide, the rate of subsidence of the vibrations will be materially increased by restarting them frequently so that the oscillations are kept up almost continuously for a considerable period. This increased rate of subsidence exhibits the well known phenomenon of the fatigue of elasticity. If the limits of strain are moderate this fatigue will not present itself but, on the contrary, should the wire be kept vibrating, the rate of subsidence of the vibrations will soon fall to a definite minimum, showing a diminished absorption of energy. The rate of subsidence will be diminished by annealing, at least in many cases, and will be increased by fluctuations of temperature. It will also be increased, but only temporarily, by a change in the longitudinal traction to which the wire is submitted. This latter effect is most marked when the load is increased, but also occurs when it is decreased. If the oscillations are kept up the rate of subsidence under the changed traction quickly falls to the same, or very nearly the same, minimum as before the change.

We need only deal here with this minimum rate of subsidence and absorption of energy, as in an engine the frequent reversal of stress will soon bring the metal nearly into this state. When this state has been obtained in a vibrating wire, or after "accommodation," as Wiedemann terms it, the law of subsidence becomes extremely simple and may be stated with practical accuracy as follows:

Law of subsidence of oscillations.

The amplitude of each oscillation is less than that preceding it in a constant proportion; or, the logarithms of succeeding oscillations differ by a constant amount.

As the energy of vibration is proportional to the square of the amplitude, the law may also be stated in the following form more suitable for our present object:

At each oscillation there is a definite fraction of the energy of the vibration abstracted.

It will be remarked that in these statements there is no reference to the period of oscillation. As the abstraction of energy due to the molecular action of the wire has been found to be independent of the velocity of change of strain, the action is in no way analogous to fluid friction, as has sometimes been supposed. Wiedemann accounts for it* by "a displacement of the molecules on one another, and at the same time also the alterations of direction of their axes, or their rotations about their center of gravity." Wiedemann's observations are fully supported by an extensive and most careful set of experiments by Mr. Herbert Tomlinson, some of whose results I will quote.†

But I must also refer to the early experiments by Sir William Thompson (Lord Kelvin), who was a pioneer in this as in so many other branches of research. Though his conclusions were not quite definite he first stated the law of subsidence of vibration and threw the greatest light on the whole phenomenon. Others have followed where he led.‡

I regret that the great series of experiments on this subject by the late Prof. Bauschinger of Munich have not been accessible, as from them we could have derived constants for the viscosity of metals vibrating transversely. From Todhunter and Pearson's History of the Theory of Elasticity, §§ 709 and 710, we learn that the behavior of rods under cross bending is very similar to that under torsion.

Without the constants for other than torsional vibrations it is impossible to assign, with any probable accuracy, the true importance of the action of viscosity; but the short treatment of the results of the action of this phenomenon in Chapter XIII, even with the true constants very doubtful, will be sufficiently instructive.

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* See Wiedemann, *On Torsion*, *Phil. Mag.*, 1880, page 97.

† See *The Influence of Stress and Strain on the Physical Properties of Matter*; by Mr. Herbert Tomlinson, B. A. *Phil. Trans. of the Royal Society*, Vol. 177, p. 801.

‡ See *Proceedings of the Royal Society*, 1865; and *Encyclopaedia Britannica*, article Elasticity.

TABLE IV.—VISCOSITY OF STEEL AND IRON.

Number of Tomlinson's experiment.	Description of wire.	Logarithm of ratio of the amplitude of succeeding vibrations.	Ratio of the amplitude of succeeding vibrations.	Ratio of the energy of succeeding vibrations = square of ratio of their amplitudes.
III	Unannealed pianoforte wire...	Mean .000351	1.0008 : 1	1.0016 : 1
VII	Same wire as Exp. III.....	Max. .000753	1.0017 : 1	1.0035 : 1
XIX	Annealed iron wire.....	Max. .005606	1.0130 : 1	1.0262 : 1
		Min. .001246	1.0029 : 1	1.0058 : 1
	Value taken in present investigation.....	.....	.....	1.004 : 1

The rise in the values from Exp. VII over those of Exp. III is entirely due to changes of temperature not exceeding 2° centigrade per hour.

The variations of value in Exp. XIX are due to sudden changes of longitudinal traction.

I shall choose 1.004 as the rate of decrease of energy, that is, each vibration loses .004 of its energy. This is rather higher than the highest value for pianoforte wire in Table IV and lower than the lowest value for wrought iron wire. The parts principally contributing to the elasticity of most gears are generally made of mild steel at the present time, which probably stands, in respect to viscosity, in this intermediate position.

Chapter V.

MOTION OF VALVE WITH FRICTION. FRICTION MUCH LARGER THAN RIGID GEAR INERTIA.

19. CHANGE OF CENTER OF VIBRATION THROUGH FRICTION AT REVERSAL.

We must now consider the problem of the motion of the valve, or any mass, actuated by an elastic gear and reciprocating between straight guides which give a constant frictional resistance always opposing the motion.

I would first observe that so long as the valve, or mass M which was supposed to be hung by a spring, has its direction of sliding

unreversed, the problem is precisely the same as if there were no resistance. If M is sliding upward, with friction, between guides, there is merely a constant force F added which acts downward and adds itself to the force of gravity of the mass. This force would draw out the spring by a distance F/f, and the constant k in equation (1), measured positive upward, will be diminished by F/f. That is, the center of vibration is lowered by this amount.

If M is sliding downward the center of vibration is raised by the distance F/f.

As a clear grasp of this fact is of the utmost importance in what follows I will dwell on it for a moment.

When there was no friction and M was hanging quite still its gravity was counterbalanced by the upward pull of the spring, the point of rest being given by equation (1) as $r = k$. When M was displaced below this point by a distance δr the elastic upward force of the spring was increased by $f\delta r$, producing an acceleration directed towards the point from which M was displaced and at which the opposing forces balanced. Similarly, if M was displaced upward from this point, gravity preponderated and the acceleration was downward. Hence a vibration about this point was maintained continuously; when M was below it and moving downward the difference of the elastic reaction and gravity abstracted kinetic energy from the mass but this was exactly restored over the same part of the path on the upward return, and the conditions at the end of each complete vibration were exactly the same as at the beginning.

Let us now suppose that M is moving with constant friction between guides which, for illustration, we may suppose to have a downward motion, constant or variable, which is always greater than the downward velocity of M. The force F is added to gravity and M could not be in equilibrium unless the strength of the elastic reaction was also increased by a quantity.

$$f\,\delta r' = F.$$

Hence the point at which M will rest has moved downward by a distance F/f. It would no doubt be impossible to realize this state of equilibrium exactly as F is not quite constant and

its variability would be increased by the impossibility of making the rubbing surfaces of quite uniform smoothness. But this we are not considering at present.

If M is now displaced by a distance ∂r below its new position of equilibrium the elastic upward reaction of the spring will again preponderate, by $f\partial r$, over gravity and friction combined, and the only change from the case of no resistance is this shifting down of the point about which the vibrations are executed.

Should the direction of sliding be reversed by the downward velocity of M becoming greater than that of the guides—which would occur for stationary guides, as for a valve sliding on its seat, at reversal of the motion of M—the point about which vibrations are executed would instantly shift to a distance.

$\partial r = F/f$ *above* the point of equilibrium for no frictional resistance. Hence the conditions of the motion change abruptly at every reversal of the frictional force, and the values of the arbitrary constants A and λ, equation (22), will have to be sought anew to satisfy the altered conditions. The effects of this abrupt change are most interesting to follow and they lead to results of great practical importance.

The most useful cases to investigate are for,

(a) The ratio $J = \dfrac{\text{Frictional resistance}}{\text{Rigid gear inertia at reversal}}$ large,

(b) The ratio $J = \dfrac{\text{Frictional resistance}}{\text{Rigid gear inertia at reversal}}$ small.

Case (a) holds in most instances for flat slide valves even of high speed engines; Case (b) in most cases for piston valves of even moderately high speed engines.

The rigid gear inertia force will be estimated from the polar valve diagram, at each extreme of the travel, by equation (6) as φ must always be zero at these points.

20. MOTION WHEN J IS LARGE.

The ratio $J = \dfrac{\text{Frictional resistance}}{\text{Rigid gear inertia at reversal}}$ large.

In Fig. 8, OX is a time base from which the valve displacements are measured. The full line curve *abc* is a portion of the valve motion deduced from the valve gear diagram; that is, it represents what would be the motion of the valve if the gear were rigid. The valve reaches one end of its stroke at *b* and we shall reckon the time t as zero at the ordinate Ob.

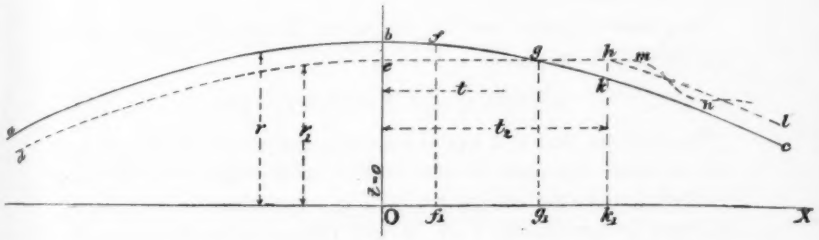

Fig. 8.

As the valve is moving outwards from OX the gear will have to overcome the frictional resistance F. This will cause the valve displacement r_1 to be slightly less than r, its value for a rigid gear, and the actual motion will be represented by a dotted line *de* very near *ab*. The exact form of the curve *de* would be got from equation (22) remembering that k is to be changed by F/f, thus,—

$$r_1 = k \pm \frac{F}{f} + \Sigma_1^{\infty} \frac{a_n}{1 - T^2/T_n^2} \cos(n\omega t - \varepsilon_n) + A\cos(\omega_s t - \lambda), \quad . . \quad (27)$$

where the sign of the term F/f and also the values of A and λ alter at each reversal of the motion.

Suppose for the moment that A is zero and that it is for this case that the dotted line *de* represents the valve motion. The maximum point *e* will fall almost at the same instant as *b*, and is represented in Fig. 8 as a point on the ordinate Ob; for the variable part of the displacement.

$$\Sigma_1^{\infty} \frac{a_n}{1 - T^2/T_n^2} \cos(n\omega t - \varepsilon_n)$$

is principally due to the longer period terms which are but little different from the rigid gear values in any reasonably stiff gear, as for them T^2/T_n^2 would be very small.

Thus this portion of the displacement is practically the same as for the full line,

$$\Sigma_1^\infty a_n \cos(n\omega t - \varepsilon_n),$$

got from equation (18).

Very near Ob, then, we may represent the actual motion very approximately by

$$r_1 = const \pm F/f + A\cos(\omega_s t - \lambda). \quad . \quad . \quad . \quad (28)$$

Thus we see that if A has any sensible magnitude there will be one or more reversals of the motion immediately before e is reached, (unless the natural period T is a considerable fraction of the time of revolution T_1.) M will then be retarded through nearly the whole of this portion of its course—that is both when approaching and receding from OX—by the friction F, which will then continually drain the energy of the vibrating system. Hence at each reversal the value of A will fall, and as the friction is large its value will be almost inappreciable at e. This is very obvious, as clearly no vibration of small amplitude, and consequently small energy, if existing nearly alone and having no powerful source of supply of energy, could perform many complete oscillations against a large frictional resistance, but would be extinguished almost immediately.

At Ob ($t = 0$) we have, from equations (27) and (18), A being zero,

$$r_1 - r = -\frac{F}{f} + \Sigma_1^\infty a_n \left(\frac{1}{1 - T^2/T_n^2} - 1\right) \cos \varepsilon_n$$

$$= -\frac{F}{f} + \Sigma_1^\infty \frac{a_n T^2}{T_n^2 - T^2} \cos \varepsilon_n. \quad . \quad . \quad . \quad . \quad . \quad . \quad (29)$$

If we call I' the inertia of the valve at any time for the displacement r_1, A being zero, we may designate the inertia at e by I_e'. Equation (27) gives,—

$$I_e' = \frac{M}{g} \Sigma_1^\infty \frac{a_n n^2 \omega^2}{1 - T^2/T_n^2} \cos \varepsilon_n.$$

(This gives I_e' positive if the curve de, at e, is concave to O, Oe being positive. Thus positive inertia tends to increase Oe. Friction will also be reckoned positive when it tends to increase Oe.)

The consequent yield of the gear is

$$\frac{I_e'}{f} = \frac{M}{fg} \Sigma_1^\infty \frac{a_n n^2 \omega^2}{1 - T^2/T_n^2} \cos \epsilon_n. \quad . \quad . \quad . \quad . \quad (30)$$

By equations (16), (17), and (21),

$$\sqrt{\frac{fg}{M}} = \omega_s = \frac{2\pi}{T}, \text{ and } n\omega = \frac{2\pi}{T_n}.$$

Hence equation (30) becomes

$$\frac{I_e'}{f} = \Sigma_1^\infty \frac{a_n T^2}{T_n^2 - T^2} \cos \epsilon_n, \quad . \quad . \quad . \quad . \quad . \quad (31)$$

and equation (29) may, therefore, be written

$$be = r_1 - r = \frac{-F + I_e'}{f}, \quad . \quad . \quad . \quad . \quad . \quad (32)$$

or, approximately

$$be = -\frac{F}{f}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (33)$$

This merely expresses again that $f\delta r$ measures the stress on the spring, and might have been written down at once. But it seemed well to deduce it formally from the previous equations.

As the crank revolves farther the ordinate ff_1 of the full line curve diminishes and the stress on the valve while in the same direction as when $t = o$, so long as ff_1 is greater than Oe, decreases. The valve therefore will remain stationary as represented by the dotted line eh parallel to OX. At the point g, where bc and eh intersect, the stress on the gear is zero; after that it is reversed and tends to start the valve back. This it cannot do until the difference hk of the ordinate of the full and dotted lines is given by

$$hk = \frac{F}{f}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (34)$$

After h the valve motion is given by equation (27) with the positive sign prefixed to the term F/f. To determine A and λ we have the two conditions,—

1st. The velocity at h is zero.

2d. As the elastic reaction is just rising above the friction F, the force starting the valve increases from zero at h; consequently at this point the acceleration of the valve is also zero.

Thus, at h putting the time $eh = t_2$,

$$\frac{dr_1}{dt} = -\Sigma_1^{\infty} \frac{a_n n\omega}{1 - T^2/T_n^2} \sin(n\omega t_2 - \varepsilon_n) - A\omega_s \sin(\omega_s t_2 - \lambda) = 0, \quad . \quad . \quad . \quad (35)$$

and $$\frac{d^2r_1}{dt^2} = -\Sigma_1^{\infty} \frac{a_n n^2\omega^2}{1 - T^2/T_n^2} \cos(n\omega t_2 - \varepsilon_n) - A\omega_s^2 \cos(\omega_s t_2 - \lambda) = 0. \quad . \quad . \quad . \quad (36)$$

Now it is to be noted that the time eh, for which the valve remains stationary, is an extremely small fraction of the time of revolution of the engine, and its representation in Fig. 8 is greatly exaggerated to render it clearly visible. Thus, we may assume that the portion of the acceleration, equation (36),

$$-\Sigma_1^{\infty} \frac{a_n n^2\omega^2}{1 - T^2/T_n^2} \cos(n\omega t_2 - \varepsilon_n)$$

is the same as $$-\Sigma_1^{\infty} \frac{a_n n^2\omega^2}{1 - T^2/T_n^2} \cos \varepsilon_n = -\frac{gI_e'}{M},$$

the value it had at e, by equation (30).

Hence from equations (36) and (17)

$$A\cos(\omega_s t_2 - \lambda) = -\frac{1}{\omega_s^2} \cdot \frac{gI_e'}{M} = -\frac{I_e'}{f}. \quad . \quad . \quad (37)$$

To determine $A\sin(\omega_s t_2 - \lambda)$ we observe first that the total decrease of the ordinate of the full line curve, from b to k is given by the sum of equations (33) and (34) as

$$Ob - kk_1 = \frac{2F}{f}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (38)$$

I being the inertia stress of the valve for the full line curve given by a rigid gear, we may designate its value at b by I_b. We have then, very approximately for a short distance very near b,

$$\frac{M}{g}\frac{d^2r}{dt^2} = -I_b. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (39)$$

From this by two integrations we get,—

$$\frac{M}{g}(r - Ob) = -\tfrac{1}{2} I_b t^2. \quad . \quad . \quad . \quad . \quad . \quad (40)$$

there being no constant for the first integration of equation (39) as the velocity is zero when $t = 0$. In the second integration the constant is determined by the condition that for $t = 0$, $r = Ob$.

At hk, $r = kk_1$, and the time eh is, from equation (40)

$$+\tfrac{1}{2} I_b t_2^2 = \frac{M}{g}(Ob - kk_1)$$
$$= \frac{2FM}{gf}$$

by equation (38); or,

$$t_2 = 2\sqrt{\frac{FM}{gfI_b}}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (41)$$

In equation (35) the part of the velocity

$$-\Sigma_1^\infty \frac{a_n n\omega}{1 - T^2/T_n^2} \sin(n\omega t - \varepsilon_n)$$

gives rise to the acceleration.

$$-\Sigma_1^\infty \frac{a_n n^2\omega^2}{1 - T^2/T_n^2} \cos(n\omega t - \varepsilon_n);$$

which, for $t = t_2$ we have just taken as

$$= -\frac{gI_e'}{M}.$$

Hence this portion of the velocity at h must be

$$= -\frac{gI_e'}{M} t_2 = -\frac{gI_e'}{M} \times 2\sqrt{\frac{FM}{gfI_b}}.$$

As I_e' does not usually differ greatly from I_b, we may with sufficient accuracy put this velocity

$$= -2\sqrt{\frac{gFI_e'}{fM}}.$$

Hence, equation (35) gives,

$$A\omega_s sin\,(\omega_s t_2 - \lambda) = -2\sqrt{\frac{gFI_e'}{fM}},$$

or

$$A sin\,(\omega_s t_2 - \lambda) = -\frac{2}{f}\sqrt{FI_e'}. \quad . \quad . \quad . \quad . \quad (42)$$

From equations (37) and (42)

$$tan\,(\omega_s t_2 - \lambda) = 2\sqrt{\frac{F}{I_e'}}.$$

As the ratio F/I_e' is large $tan\,(\omega_s t_2 - \lambda)$ must be comparatively large, and $sin\,(\omega_s t_2 - \lambda)$ not far from unity. Thus equation (42) gives, very approximately,

$$A = \frac{2}{f}\sqrt{FI_e'}.$$

The maximum stress arising from the part of the displacement, $Acos\,(\omega_s t - \lambda)$, is

$$\frac{M}{g}A\omega_s^2 = 2\sqrt{FI_e'}, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (43)$$

and it acts along with the friction and inertia in, approximately, the short time $T/4$ after h is passed.

Thus, the whole stress, instead of being,

$$F + I_b,$$

as usually taken (if, indeed, the inertia is estimated at all) is now seen to be

$$F + I_e' + 2\sqrt{FI_e'}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (44)$$

As F is not ascertainable with much precision, and as the inertia is small compared with it; also, since the effect of elas-

ticity in changing I_b into I_e' is difficult to estimate, we may write, calling S the total stress of the valve,

$$S = F + I_b + 2\sqrt{FI_b}. \quad . \quad . \quad . \quad . \quad . \quad (45)$$

21. NUMERICAL RESULTS—GREATLY INCREASED INERTIA.

The following table exhibits the very considerable importance of the third term.

TABLE V.

F	I_b	$2\sqrt{FI_b}$	S.
1	1/4	1	2 1/4
1	1/9	2/3	1 7/9
1	1/16	1/2	1 9/16
1	1/100	1/5	1 21/100

The effect is seen to be a very important one, very sensible even when the inertia is only 1 per cent. of the friction. It justifies in this case the comparatively high apparent factors of safety which are commonly used.

That this term should have been completely overlooked, if, indeed, it would prove after a proper search that it had been, is most surprising; as, on slight reflection, it becomes obvious that some such effect must always occur. That the gear must be under strain when the valve with much friction stops, is clear. Hence the momentary but finite rest of the valve. When *h* Fig. 8, is reached we should have

$$S = F + I_e'$$

only for a valve movement given by the dotted curve *hl* which is the continuation of *de* moved out $2F/f$ from *OX*. But this would require a sudden finite change of motion at *h*. Hence some such path as *hmn* must be described with a corresponding increase of inertia.

It is to be carefully observed that equation (45) is not changed by altering the stiffness of the gear. For a stiffer gear *be* and *eh* would be diminished, and consequently the velocity at *k*, for the full line curve, would be smaller, thus jerking the valve less at the start; but this is balanced by the shortening of the natural period, and thus the value of the third term is unaffected.

The small vibration of time T in the motion *hmn* will be increased or diminished from *h* onwards according to the relation of its epoch to that of the synchronizing term of the rigid gear. But this effect we need not examine here as we know that on the valve approaching the other end of its travel it will be damped out.

As will be readily understood after the energy transfer curves, Fig. 21, have been explained, in adopting the approximate equation (33) instead of equation (32) we make nearly the worst supposition possible as to the vibration $A\cos(\omega_s t - \lambda)$ remaining when *e*, Fig. 8 is reached. For the foregoing demonstration shows that the larger *be* the larger will be the resultant stress on restarting the valve. $A\cos(\omega_s t - \lambda)$ will increase *be* most if it has its maximum displacement toward OX when *e* is reached. In that case Fig. 21 shows that if its inertia is much greater than I_e' there will be considerable absorption of its energy of vibration. Equation (33) would be exact if its maximum displacement toward OX occurred at *e* and its inertia was equal to I_e'.

REHEATERS IN MULTIPLE-CYLINDER ENGINES.*

By R. H. Thurston, Ithaca, N. Y.

Reheaters in multiple-cylinder engines have been employed by various designers and builders from a very early period in their history. Perhaps the first to use this device was John Bourne, who tells us that he employed it in 1859. In later years, Mr. Cowper, the distinguished English hydraulic engineer, employed the device, and, in this country, as early as 1869, it was used by the late Henry R. Worthington, in his Philadelphia pumping engine and on many later engines. It was used by Mr. Corliss, and also by Mr. Leavitt on his Calumet and Hecla Mining Company engines (1874).

The first mention of this apparatus in engineering literature, so far as the writer has observed, occurs in the little book by John Bourne, "Recent Improvements in the Steam Engine," in which are contained so many minor practical points of value in connection with this form of prime motor. He speaks of it as a "superheater," showing his precise understanding of its place and purpose, and he evidently perceives its use to be solely that of giving a better working substance in the low-pressure element of the compound engine, to which, only, he has applied it; the triple and quadruple-expansion engines not having at that time come into use and not being required at the comparatively low steam pressures then in vogue. Bourne says:

"In some cases, where double-cylinder engines are employed, a superheater is introduced between the high-pressure cylinder and the low-pressure one so as to replace the heat lost by radiation and by the production of power in the high-pressure cylinder. This arrangement, introduced by me into a steamer, about 1859, was exemplified in various engines shown at the International Exhi-

* Paper presented at the Cincinnati meeting, May, 1900, of the American Society of Mechanical Engineers.

bition in 1862, and promises to be very serviceable in cases where a small consumption of fuel is an object of paramount importance." *

Bourne thus recognized its exact function and also states clearly his understanding that the loss of heat which it is intended to counteract comes, in part, from the performance of work in the high-pressure cylinder. He very probably also understood, although he does not so state, that the superheating sought is a remedy for effects in reduction of temperature of the low-pressure cylinder by that and other methods of heat waste provocative, in that cylinder, of initial condensation. †

About 1870, H. M. S. *Briton* was fitted with compound engines of 57 and 100 inches diameter of cylinder and 33 inches stroke of piston, with intermediate reservoir and reheater of the type introduced by Mr. Cowper years before. Steam was carried at 60 pounds, and the engines developed 2,018 indicated horsepower for six hours on her trial, consuming 1.98 pounds of fuel per indicated horsepower, hour. Surface condensers and reheating contributed to this result. At reduced powers, still better figures were reported. The relative economy of the then usual classes of engines for marine purposes was reported at the time, thus: ‡

RELATIVE ECONOMY OF MARINE ENGINES—RENNIE.

"Improved"	2	pounds coal per indicated horsepower, hour.
Simple, with surface condensers	3.5	pounds coal per indicated horsepower, hour.
Simple, with jet condensers	4.5	pounds coal per indicated horsepower, hour.
Non-condensing engines	6	pounds coal per indicated horsepower, hour.

In a discussion of a paper by Mr. Grantham "On Ocean Steam

* "Recent Improvements in the Steam Engine," John Bourne, p. 15; also "Reheating," John Bourne, 1859. Thurston's "Manual of the Steam Engine," vol. i, p. 888, footnote.

† This anticipation of modern ideas and practice is but one of many exemplifications of the fact that Bourne was far in advance of his time and foresaw and often applied practically nearly all the finer developments of later engineering in this field.

‡ "On Improved Compound Engines," by G. B. Rennie, Esq., March 30, 1871: Trans. Brit. Inst. N. A., 1871.

TEST OF 21 × 42 × 13.5 × 40 CROSS COMPOUND AT NEWTON, MADE JUNE 9, 1892, TO DETERMINE THE VALUE OF THE REHEATER.

Duration of test	Two hours.	Two hours.	Two hours.
Total number of revolutions	4,622	4,584	4,616
Pressure of the atmosphere	14.6 lbs.	14.6 lbs.	14.6 lbs.
Pressure by gage at the throttle	116.0 lbs.	119.25 lbs.	119.0 lbs.
Absolute pressure at the throttle	130.6 lbs.	133.85 lbs.	133.6 lbs.
Absolute press. in the reheater	Not in use.	*135.71 lbs.	*39.52 lbs.
Total press. per sq. in. on pump	119.54 lbs.	118.08 lbs.	113.70 lbs.
Temp. of main feed at engine	131.27° F.	126.30° F.	125.50° F.
Temp. of main feed at boiler	127.20° F.	123.55° F.	124.00° F.
Temp. reheater drain at engine	Not in use.	344.35° F.	263.85° F.
Temp. reheater drain at boiler	Not in use.	290.35° F.	259.85° F.
Weight of steam through cyls.	7,586 lbs.	6,832 lbs.	7,185 lbs.
Weight of steam through reheater	Not in use.	440 lbs.	119 lbs.
Weight of water fed to boiler	7,586 lbs.	7,272 lbs.	7,304 lbs.
x = dry steam per lb. of mixture	0.993 lbs.	0.993lb.	0.993 lb.
r for pressure at the throttle	869.16	867.8	867.9
q for pressure at the throttle	318.76	320.7	320.6
q_0 for main feed at the engine	98.37	94.4	93.7
q_2 for reheated drain at do.	Not in use.	315.4	233.0
$(xr+q-q_0)$ heat above 32° per lb. in cylinders	1,083.5	1,088.0	1,088.7
$(xr+q-q_2)$ heat above 32° per lb. in reheater	Not in use.	867.0	949.4
Total B.T.U. used in cylinders	8,219,431	7,433,216	7,822,309.5
Total B.T.U. used in reheater	Not in use.	381,480	112,978.6
Total B.T.U. used by engine	8,219,431	7,814,696	7,935,288.1
Revs. times the pump press.	552,513.88	541,278.72	524,839.2
Ratios of do.	1	0.9796	0.9499
Ratios of feed water used	1	0.9586	0.9720
Revs. × pump press. ÷ B.T.U.	0.06722	0.06926	0.06614
Ratios of do.	1	1.0304	0.9839
Revs. × pump press. ÷ feed	72.8333	74.4332	71.8564
Ratios of do.	1	1.0219	0.9866

REMARKS.

During the tests all of the water fed to the boiler was pumped by the main feed pump attached to the engine, this water being carefully weighed as on previous tests. The water in the boiler was at the same height at the beginning and end of tests, and was practically at this height throughout. When the reheater was in use the water of condensation was collected and weighed. Steam was supplied to the reheater by a pipe leading from the main pipe near the engine.

In this engine steam enters the top of the jacket surrounding the high-pressure steam cylinder, from which jacket it passes through Corliss valves at the top into the cylinder. The steam after being used in this cylinder escapes through valves of the same type at the bottom into a tubular reheater, the reheating steam passing through the tubes. From the reheater the steam enters the bottom of the jacket surrounding the low-pressure steam cylinder, from the top of which it enters the cylinder through Corliss valves, escaping at the bottom into a receiver containing a coil of copper pipe, through which coil the water from the main feed pump passes on its way to the boiler. From the last receiver the steam passes through a jet condenser. In the normal running of the engine the water of condensation in the reheater is pumped to the boiler by a pump attached to the engine. All the heads are jacketed with the steam entering the main jackets. All jacket water was trapped off and thrown away. The piston rods, 4 inches in diameter, pass through one head of each steam and pump cylinder. The main steam and pump cylinders are horizontal. The air pump is vertical and single acting; the feed pumps are vertical, single-acting plungers, all attached to the engine. Carefully conducted calorimetric tests show the steam at the engine to contain 0.7 per cent. moisture. All the instruments and gages were carefully tested. A mercurial barometer was used.

ALBERT F. HALL, M. E.

* These gages were too small for accurate readings.

Navigation," before the British Institution of Civil Engineers, in 1869,* Mr. Cowper stated:

"It was many years since he had introduced, with good results, the steam-jacketed reservoir between a high-pressure and a low-pressure engine, with the cranks at right angles. Seven years before, he had improved upon this and had made a steam-jacketed reservoir of peculiar construction. This was so arranged that the high-pressure steam, after having been expanded in the high-pressure cylinder, and thus become low-pressure steam, entered the steam-jacketed reservoir without being heated; but the steam, on leaving the reservoir and entering the low-pressure cylinder, was heated by a peculiar arrangement of the hot steam-jacketed side of the reservoir."

One of the few trials of engines supplied with reheaters which have afforded useful data is that of Mr. Albert F. Hall, a discussion of which appears in the JOURNAL OF THE AMERICAN SOCIETY OF NAVAL ENGINEERS, November, 1893. Its tabulated data are here reproduced with the appended remarks. In the first of these trials the reheater was out of use, and in the third employed but one-half as much steam as in the second. It is seen that the results of its use in the third trial are identical with those of the first, within the limits of accurate observation, although a loss of 1 per cent. was indicated; while the second trial showed a gain by the use of reheater of about 3 per cent., employing double the amount of the steam in the apparatus that was used in the last trial.

Assuming that we may rely upon the indications of these trials, notwithstanding the small differences reported, we are justified in the conclusion that unless the reheater is made effective in superheating it is best not to employ it at all, relying in such case upon separating apparatus. It is worse than useless if superheating is not insured by its employment.

"The Engineering Record," in its issue of April 16, 1898, publishes the results of an engine trial by Mr. Barrus, of which the following are the essential data:

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* Minutes of Proceedings of British Inst. C. E., vol. xxix, 1869–70.

DATA AND RESULT OF FEED-WATER TESTS OF PAIR OF TANDEM COMPOUND ENGINES (NO. 2 AND NO. 3 ENGINES), GROSVENOR-DALE COMPANY.

Nominal receiver pressure....	5 pounds.		9 pounds.		13 pounds.		13 pounds.	
Jackets.....	Jackets on.		Jackets on.		Jackets on.		Jackets off.	
Date.....	Oct. 28, 1897, A. M.		Oct. 28, 1897, P. M.		Oct. 29, 1897, A. M.		Oct. 29, 1897, P. M.	
1. Duration, hours.....	5		5.067		5.033		4.5	
2. Total weight of feed water consumed, pounds.....	47,945		50,632		49,295		44,221	
3. Weight of water consumed per hour corrected for leakage and moisture, pounds	9,399		9,802.5		9,604.4		9,616.9	
4. Indicated horsepower developed by H.P. cylinders, I.H.P.....	347.63		342.48		323.48		350.79	
5. Indicated horsepower developed by L.P. cylinders, I.H.P.....	371.22		398.84		415.29		374.41	
6. Indicated horsepower developed by whole engine, I.H.P.....	718.85		741.32		738.77		725.2	
7. Average pressure in steam pipe near engine, pounds	144.9		144.1		143.8		144.1	
8. Average pressure in receivers, pounds	4.8		8.6		12.2		12.3	
9. Average vacuum in condensers, inches.....	26.4		25.2		25.0		24.9	
10. Average revolutions per minute, revolutions.....	77.45		76.63		76.86		78.89	
11. Mean effective pressure H.P. cylinders, pounds...	61.12		60.82		57.29		60.53	
12. Mean effective pressure L.P. cylinders, pounds...	9.76		10.6		11		9.66	
13. Dry steam consumed per I.H.P. per hour, pounds..	13.08		13.22		13		13.26	
14. Pressure in steam pipe, pounds	145		144.5		145		144.5	
15. Receiver pressure, pounds	4.8		8.5		12.3		12.8	
	H.P. Cyl.	L.P. Cyl.	H.P. Cyl.	L.P. Cyl.	H.P. Cyl.	L.P. Cyl.	H.P. Cyl.	L.P. Cyl.
16. Initial pressure above atmosphere, pounds.....	140.2	6.4	140	9.7	139.7	13	139.9	11.7
17. Cut-off pressure above zero, pounds.....	138.1	16	140.1	19.5	140	22.4	140	21.4
18. Release pressure above zero, pounds.....	38.5	5.7	40.3	5	41.3	5.6	44	5.2
19. Compression pressure above zero, pounds...	35.3	4.2	37.5	4.5	42.5	4.1	42.2	4.6
20. Mean effective pressure, pounds.....	62.62	10.2	60.64	10.69	57.64	11.07	60.3	9.58
21 Back pressure at midstroke above or below atmosphere, pounds.....	+6.2	−12.5	+10.3	−12.5	+13.8	−12.2	+14.1	−12.2
22. Proportion of forward stroke completed at cut-off.....	.256	.357	.265	.283	.268	.236	.293	.213
23. Proportion of forward stroke completed at release.....	.962	.987	.966	.973	.962	.977	.953	.978
24. Proportion of backward stroke uncompleted at compression	.07	.06	.076	.053	.075	.064	.085	.06
25. Steam accounted for at cut-off, pounds.....	8.32	10.44	8.46	9.93	8.48	10 23	9.75	8.71
26. Steam accounted for at release, pounds.....	8.97	10.43	9.14	10.09	9.26	10.62	10.27	9.79
27. Proportion of feed water accounted for at cut-off...	.636	.798	.64	.751	.652	.787	.735	.656
28. Proportion of feed water accounted for at release...	.685	.797	.69	.763	.712	.817	.775	.738

DIMENSIONS OF ENGINE.

Diameter of right high-pressure cylinder, inches	$14\frac{3}{32}$
Diameter of left high-pressure cylinder, inches	$14\frac{1}{16}$
Diameter of high-pressure rods, inches	$3\frac{1}{4}$
Stroke, high-pressure cylinders, feet	4
Clearance, high-pressure cylinders, per cent	2
Horsepower constant 1 pound mean effective pressure, 1 revolution per minute each high-pressure cylinder, horsepower	.03673
Horsepower constant 1 pound mean effective pressure, 76 revolutions per minute each high-pressure cylinder, horsepower	2.79
Diameter of right low-pressure cylinder, inches	$36\frac{3}{32}$
Diameter of left low-pressure cylinder, inches	$36\frac{3}{32}$
Diameter of low-pressure rods, front end, inches	$3\frac{1}{4}$
Diameter of low-pressure rods, crank end, inches	$4\frac{1}{2}$
Stroke low-pressure cylinders, feet	4
Clearance, by drawing, per cent	2.5
Horsepower constant 1 pound mean effective pressure, 1 revolution per minute each low-pressure cylinder, horsepower	.24555
Horsepower constant 1 pound mean effective pressure, 76 revolutions per minute each low-pressure cylinder, horsepower	18.66
Ratio of volumes	6.69 to 1
Heating surface in two reheaters, square feet	220
Heating surface in feed-water heater, square feet	172

With jackets in action the consumption of steam was divided thus: working steam in cylinders, 89 per cent., jacket steam, 11 per cent.

The editor, commenting upon this exhibit, says:

"An economy of 2 per cent. in favor of jacketing and reheating, which was the amount shown by these trials, is hardly worth considering when we take into account the extra complication and possible increase in the danger of breakage which are incident to their use. In an engine of the size noted, say 650 indicated horsepower, the consumption of coal per week of 60 hours' run is in round numbers, 25 tons; or, for a year's run, 1,300 tons. At $4 per ton, this represents an expenditure for fuel of $5,200 per annum. A saving of 2 per cent. on this sum is $104 per year, and this is 10 per cent. of $1,040. The extra investment for the installation of jackets and reheaters, which the saving warrants, cannot, therefore, under these circumstances, be much over $1,000. It is probable that the addition of the jackets and reheaters to the engine in question required the expenditure

of at least the sum just named; so that the return indicated is just barely sufficient to pay the fixed charges on the investment. There is, however, to be taken into account the heat of the jacket water, which, if returned to the boiler, produces some advantage. In the tests in question, the temperature of the feed water was raised by the admixture of the hot water from the jacket traps 25 degrees. This, in itself, represents a saving of about 2 per cent.; but on the basis of calculation above noted, it gives only about $100 saving in a year; and such an amount is hardly worth having when the complication and possible danger incident to jacket use is properly valued.

"The result obtained on this comparison is precisely similar to that shown on a similar trial, made on an engine at the Grosvenordale Mills in 1894.* In that case, with a tandem compound engine of about the same power as the one under present consideration, and of the same general proportions, but having two cylinders instead of four, the consumption of steam per indicated horsepower per hour with the jackets and reheaters in use, was 12.45 pounds. When the jackets and reheater steam were shut off, the other conditions remaining precisely the same, the consumption was 12.69 pounds per indicated horsepower per hour, the difference being 2 per cent. This goes to show that the comparative small effect produced by the jackets and reheater is not that of an insolated case, but it is one which is borne out by repeated investigations on different engines of the same type.

"If in the recent tests we analyze the diagrams and compare one set with the jackets in use, with another set with the jackets not in use, it appears that there is a very marked difference; and that the failure of the jacket to show a greater advantage is not due by any means to inaction. The noticeable feature in the operation of the jacket is that it increases the work done by the low-pressure cylinder. In the two comparative trials referred to in the abstract of the report, namely, the tests of October 29th, forenoon and afternoon, the horsepower developed by the low-pressure cylinder with the jacket in use, was 29 per cent. more

* The "Engineering Record" of November 3, 1894.

than that developed by the high-pressure cylinder. When the jackets were shut off, the increase in the power developed by the low-pressure cylinder was only 7 per cent. of the power developed by the high-pressure cylinder. Furthermore, the steam accounted for by the diagrams when the jackets were in use was much larger in the low-pressure cylinder than in the high-pressure cylinder; while, on the contrary, when the jackets were shut off, the steam accounted for was much less in the low-pressure cylinder than in the high-pressure cylinder."

Mr. Carl Sulzer, in a letter addressed to Mr. Hamilton Hill, M. A. S. M. E., and printed in the *Transactions* of the American Society of Mechanical Engineers, in 1897,* remarks:

"We have tried the heating of all jackets by direct steam, also the heating of tubular receivers, but have found no advantage in so doing." And in a letter to the writer, Sulzer Brothers say: "We do not deny that a slight gain in economy, may, in certain cases, result from tubular reheaters; but this economy is, in all cases, made up by the interest on the increased cost of the whole plant."

Mr. Charles H. Manning, who has a large experience in steam-engine operation in cotton mills, and where economy of steam and fuel is important, states in a letter to the writer:

"I am a believer in the reheater, and know that in our own vertical triple [Amoskeag Mills, Manchester, N. H.] we receive much benefit from them; but we have not been able to get exact figures."

Mr. Richard Rice says:†

"As to a value of a reheater which is sufficient to superheat the steam—for instance, like the engine at the Washington Mills, where we are getting from 50 to 60 degress of superheat—I am not ready to form an opinion based upon any scientific facts; but I believe from what I have seen that it is of value."

Mr. E. D. Leavitt, who has had large experience, says:*

"My experience with reheater has simply proved that they are of sufficient value to warrant their use.

* *Trans. A. S. M. E.*, 1897, p. 812.

† Correspondence.

Mr. A. K. Mansfield writes: *

"We have been using reheaters to a considerable extent, and have recommended their use with the belief that they are of somewhat more advantage than the steam jacket, which we like to avoid in commercial engines, as they add complication and expense."

Mr. Edwin Reynolds says: *

"As a matter of fact, the best results we have ever obtained have been on engines where these reheating receivers have been used; but to what extent they affected the results we are unable to say."

Mr. Charles C. Worthington writes: *

"We have made enough tests in the direction of doing away with reheating to determine positively that the more reheating we get the better the engine works . . . . the stroke is more absolutely uniform the drier we make the steam. . . . . The more we do away with the disturbances of entrainment and irregular condensation the nearer we come to the ideal conditions, and superheating and reheating are in proper line for this. We find that the engine is sensitive to any attempt to reduce the pressure in the jackets or the reheater, and even if only done in the low-pressure cylinder and the last reheater, and our observations seem to indicate—although they have not been made with as much care as the importance of the subject demands—that a loss of economy follows any such attempt."

The Sibley College quadruple-expansion experimental engine, of which an account has already been given in the *Transactions* of the American Society of Mechanical Engineers,† is fitted with reheaters, and although operated at 500 pounds pressure and upward, and although the superheating effected is but moderate, it is supposed that a sensible amount of gain is due to their use, and that the reported efficiency of the engine, a consumption of less than ten pounds of steam, of below 11,000 British thermal units,

* Correspondence.

† "The Promise and Potency of High-pressure Steam," R. H. Thurston, *Trans. A. S. M. E.*, vol. xviii, 1896, No. DCCXVIII, p. 160.

per hour per horsepower, comes in some degree from this construction.

The triple engine of the Massachusetts Institute of Technology, refitted with reheaters of variable effective area of reheating surface, is reported to have given the following results in preliminary trials:

With reheaters out of use and developing 88 to 90 indicated horsepower, from 15.95 to 16.03 pounds of steam per horsepower per hour passed through the engine. With 84.93 indicated horsepower and with one reheater in use, the consumption was 15.87. With 75 and 95 horsepower developed, the two reheaters being in use, 14.9 and 14.7 pounds were used. With 105 and 107 indicated horsepower developed, the consumption was 14.7 and 14.5. Cylinder jackets were not in use. Net saving, 0.08.

The triple-expansion experimental engine of the Sibley College laboratory shows very little if any advantage in the use of the reheaters in conjunction with the jackets on the cylinders, but there is an evident more or less complete substitution of the one device for the other where used alternately, and the required improvement in quality of steam to insure the exceptional efficiency of that engine noted under its most favorable conditions of operation is approximated with either arrangement. Preliminarily well-dried steam passing through a superheater is certainly an effective system of improving efficiency.

Gain by use of reheaters properly applied is usually considered by those builders who have experimented with them as undoubted, but its magnitude is still a matter of question, and this question cannot be fully and definitely settled until direct experiment shall have afforded a direct measure and possibly revealed a law. It may, however, be probably roughly estimated from known facts bearing upon the case. It is certain that where producing superheating, cylinder condensation in the cylinder receiving steam through them will be reduced. It is known that, with large engines, such as are commonly fitted with these adjuncts, a superheat of 80 degrees to 100 degrees Fahrenheit will reduce this waste to an insignificant quantity. It is indicated by reference

to experiments with steam jackets that where the latent heat of steam of boiler pressure is employed in the reduction of this waste each pound of such steam so employed saves on an average considerably more than its own amount, and general experience would seem to show that, in the class of engines here studied, the net result should be expected to be a net saving of from two to four or five pounds of steam worked in the engine for each pound thus expended, averaging in actual reported practice probably not far from three. The first datum would show that 10 per cent. added heat in the form of superheat of originally "dry and saturated" steam saves about 20 per cent. of the total expenditure in the engine. The latter datum indicates that 10 per cent. should in this manner save 30 or 33 per cent. where cylinder condensation attains any greater magnitude than this proportion of total steam supplied. But reheaters on a compound engine can only affect the low-pressure cylinders and alone should be expected to save perhaps one-half these proportions, probably one-half the cylinder-condensation loss, which is in such engines usually not far from 20 per cent. in compound and 15 per cent. in good triple-expansion engines. A net 10 per cent. in the one and 5 per cent. per reheater in triple and quadruple-expansion engines should not be too high a preliminary estimate. Moderate superheating would presumably precede the delivery of the boiler steam to the high-pressure cylinder and this should, with the reheaters, insure a gain of fully 20 per cent. in the compound and 12 to 15 in the other machine.* Of the total gain not far from one-third should be derived from the reheaters. Exact test only can fully determine such questions and give the engineers a sufficient basis for contracts.

Reheaters have usually been employed both as separators and drying apparatus, and as superheaters, being introduced into the steam pipe, or the steam connection between cylinders, and supplied with high-temperature steam, or with heat from the flues or smokestack in sufficient amount to give them the desired effi-

* "Theory of the Steam Jacket," R. H. Thurston, *Trans. A. S. M. E.*, June, 1894, p. 65. "Superheated Steam," R. H. Thurston, *Trans. A. S. M. E.*, May, 1896, p. 69.

ciency.* If properly arranged, complete separation will be first effected, and the steam thus dried will be then raised to a temperature considerably in excess of that of the saturated steam coming into the reheater. Every good reheater is, in fact, a superheater for the next succeeding cylinder. The degree of superheat secured is usually, in the best cases, not less than 50 degrees Fahrenheit (28 degrees centigrade) and is sometimes considerably more, while more common experience indicates simply the more or less effective separation of water from the steam, and comparatively little gain except such as comes of the reduction of the quantity of moisture present at the point of cut-off. It is, however, very generally the fact that the most striking successes in the production of exceptional economy in the use of steam are exhibited by engines fitted with superheaters between cylinders, and thus securing proper and ample "reheating." Both dispositions of the reheater—taking heat from high-pressure steam and taking it from the chimney flue—are in use, and both have given good results. Choice between them will evidently be determined by the temperature of the discharged gases at the boiler, the available temperature of prime steam, and the arrangement of the engines and boilers in matters of detail. Every designer must settle this matter for himself, having in mind the conditions controlling the case in hand.

The Method of Action of the intermediate "reheater" in multiple engines would seem to be very simple, and the principles controlling its employment equally so. Its office is precisely that indicated by Bourne. Its effective use depends upon the efficiency with which it can perform that office. It is valuable in proportion to its effectiveness as a superheater, and as that only. In the many cases in which it has been found of no use it will be observed that its action results in no important drying or superheating, even where it does not allow steam

* "Superheating may be applied to the intermediate pipes or receivers of compound, triple, or quadruple-expansion engines. This may be conveniently effected by means of pipes traversing the steam pipes or receivers, and exposing heating surface, the heat being supplied by live steam of higher temperature. As in steam jacketing, the operation is attended with greatest benefit when applied to the last cylinder in a series."—*Sutcliffe.*

to pass into the succeeding cylinder wet. In those instances in which it has proved itself an economical device, according to the testimony of its designers and users, it is also to be noted that superheating in a very marked degree accompanies its action.

The reheater in a current of wet steam, where unequal to the task of drying and markedly superheating that steam, cannot be expected to prove economically useful, since both theory and repeated experiments show that—in engines in which the ratio of expansion in any single cylinder is small, at least—no observable increase in efficiency is consequent upon any variation of quality of the steam supplied, when that quality does not exceed unity sufficiently to permit reduction of initial condensation by preliminary and sensible increase of temperature of the cylinder wall and one resulting in decided decrease of the internal wastes of the system. Where the area of the heating surfaces of the reheater and the quality of the steam entering it are such as to permit drying and superheating, the effect of this latter action must be balanced against the cost of the former; thus determining a net result which, if indicating a surplus of heat applicable to superheating the current of steam, will have an economic result largely in excess of the value of the energy stored, as superheat, in this excess.

The duty of the reheater may thus be reckoned as the sum of two distinct actions: the drying of wet steam and superheating of the steam thus dried. For the first, it must be competent to supply the amount of heat, per unit weight,

$$Q = x\,l;$$

x being the quality of the steam entering the reheater and l its latent heat per unit of weight. The area of heating surface and the temperature-head producing flow into the drying steam, as it passes into and through the reheater, must be sufficient to furnish this amount and also the quantity, per unit weight,

$$Q^1 = 0.48\,(T_2 - T_1),$$

required to insure the demanded superheat for each unit weight of steam flowing through the reheater.

Temperature head in ample measure is thus at once seen to be an essential prerequisite to the successful use of this apparatus. This means that ample section of steam pipe between the boiler and the reheater must be secured to insure full boiler pressure in the reheater, and that where, as is now coming to be not unusual, boiler pressure is carried largely in excess of steam-chest pressure at the engines, the full temperature and pressure head thus available should be secured at the reheater. It is easy to reduce superheating, if desired, under such conditions; but it is not easy to increase it when deficient. The wider the range of this temperature head, $T_2 - T_1$ the less the area of reheating surface demanded for a stated amount of work in drying and superheating the steam. As this head is necessarily comparatively small, the area of this appurtenance to the multiple-cylinder engine, to be effective, is as necessarily comparitively large.

The area of surface in the reheater is thus proportional to work demanded, to the reciprocal of the temperature heads, and to the resistance to heat flow through the metal walls of the reheater. It is evidently useless to adopt an area insufficient to dry, or even sufficient to simply dry, the steam passing through it; its area must be sufficient to insure some superheating and, this point attained, the heat expended will bring a return of several times its value in all ordinary cases. This gain, as already remarked, must be charged, however, with the cost of drying quite to the point of converting the wet steam entering the reheater into thoroughly dry steam.

Since the receiver cannot act as a reheater effectively, or produce any gain, apparently, unless by superheating, it is evident that all moisture should be taken out of the steam before it enters the reheater. A separator-receiver, capable of perfect drainage, between the small cylinder and the reheater, is thus, in many cases, likely to prove an advantage.

The use of the reheater in an engine having two or four cylinders in series, where it receives its heat from the chimney gases, has this peculiar advantage, in addition to that of employing heat otherwise wasted but which might be utilized by the extension of the heating surface of the boiler: the heat absorbed from the

passing gases may be taken out, more completely than it would otherwise be, through the series action of the receivers themselves when acting as reheaters. That taking heat from the flue for use between the high-pressure and the intermediate cylinder finds the available temperature head comparatively large, and small areas of reheating surface are needed for a stated amount of work. The reheater between the low-pressure and intermediate cylinders finds a lower temperature in the flue, but its own steam has, meantime, assumed a much lower pressure and temperature than that in the first reheater, and thus the available temperature range for heat transfer becomes comparatively considerable. With very high steam pressures, and a correspondingly large number of cylinders in a series, this peculiarity and economic advantage of the series engine, fitted with reheaters of sufficient area of surface, becomes, it may be fairly presumed, an item to be considered. The cost of maintenance of these drying and superheating reheaters in the flue must, however, be always seriously considered.

The advantage of superheating is always the greater as the engine without that accessory is uneconomical. The larger the amount of cylinder condensation and the greater the losses, exterior and interior, the greater the effect of any given amount of superheating. The same statement will hold of the use of reheaters: the more wasteful the engine without them and the more effectively they superheat, the larger the gain by their use. As with superheaters of the common sort also, if they do not actually superheat they are useless, since the drying of the steam may then be much better effected by the use of the simple separator. All superheaters and all reheaters should be designed in anticipation of this contingency as a possibility, and all should be fitted up with ample drainage arrangements, whether effectively superheating in regular operation or not. The purpose of the steam jacket, the superheater, and the reheater is the same with all, and they all necessarily act in a similar manner: first drying the steam, if wet, then giving it so much of superheat as to sensibly reduce the amount of initial condensation during the first stages of the action of the fluid in the engine. Whichever is employed, or if

all are used, the use of a system of separation as a preliminary to the superheating is economically desirable, as re-evaporation of the entrained moisture is best performed in the boiler to which the separator should send it directly and without loss of heat en route. A superheating reheater is the desirable accessory, and anything short of a measurable superheat is undesirable, since the work thus performed by the complicated apparatus can be better done by one less intricate and less costly in construction and maintenance.

Conclusions, probably definite, and sufficiently accurate for practical application, may be drawn from what is already known of this subject, although we are obliged to admit that direct experiment has done little to aid us in their deduction. It is tolerably certain that a reheater, wherever and however employed, can be of little if any value in improving the thermodynamic action of the engine if it has not sufficient power to produce some superheating, and it is no less certain that a properly proportioned and placed reheater will be of comparatively small importance if supplied with wet steam, and its value will be less as the amount of the moisture in the steam entering it is the greater, up to the point at which it ceases to produce superheating of the steam entering the succeeding cylinder.

The proper construction of the reheater is thus dependent upon the following principles:

(1) A reheater should be given such area of heating surface as will insure, under the circumstances of its operation, at least moderate superheating.

(2) The reheater must, to insure proper action, be supplied with perfectly dry steam, and must usually be accompanied by a separator out of which it may take such steam, or it must itself act as a separator.

(3) To be thoroughly effective, the reheater should take steam immediately from a separator, and should deliver its own charge directly into the succeeding cylinder, thus permitting no opportunity for loss of heat and production of saturation before the charge is fairly introduced into the next steam cylinder. The

separator may probably often be constructed in one piece with the reheater, with advantage in this respect.

(4) Those conditions which control in the application of the superheater apply equally well to the case of the reheater. The latter is properly a superheater placed between cylinders in series; otherwise it becomes simply a separator.

(5) With properly proportioned and adjusted superheating and reheating apparatus, multiple-cylinder engines should gain quite considerable economy, and even enough, where high efficiency is demanded, to make their employment financially desirable.

SYSTEMS AND EFFICIENCY OF ELECTRIC TRANSMISSION IN FACTORIES AND MILLS.*

By William S. Aldrich, Champaign, Ill.

The recent progress in the use of electricity for the transmission of power over short distances has developed a new industry. It bids fair to rival in magnitude and usefulness the field of long-distance transmission, much earlier developed, and now almost exclusively held by electricity. As applied to factories and mills electricity is simply a means to an end, which is primarily the transmission of power over quite short distances, from 50 to 500 feet, and within one building or a group of buildings. Upon entering this new field it has had to contest every inch of its progress in competition with long established usage, in order to displace the unweildy and unsightly power transmissions by shafting, belting and rope drives. In almost all cases of new manufacturing plants, however, the features of electric transmission have received thorough consideration, resulting in many factory installations in which this system is exclusively used.

Some manufacturers have hoped that electricity would solve all of the problems, and at once, upon its introduction into their establishments; others have known it would be of no use from the beginning. There are many factories and mills in which the introduction of electricity for power transmission will not pay, under existing conditions; there are more establishments in which it would pay, in which an investment in electric transmission would prove to be a dividend-paying investment. No general rules can be laid down. Each must be carefully examined, and and a most thorough preliminary survey made of all the conditions and requirements.

In minor ways, electricity has proven to be a very useful ser-

* Paper presented at the Cincinnati meeting (May, 1900) of the American Society of Mechanical Engineers.

vant to have around in manufacturing work, not only for lighting the establishment, but for many lines of work requiring small but easily-controlled heat supply. Certain small industries have been completely revolutionized, and new lines of manufacturing work made feasible and profitable by the use of electricity.

Each year more and more machinery is thrown into the scrap heap to make room for new and improved types, yet antiquated and costly methods of transmitting power are left in service. Modern methods of workshop production, in which machine tools are worked up to the power limit, are seen to be sought after, side by side with belted transmission of power, which makes it quite impossible to realize fully the value of the new method, by not giving the operator the utmost freedom of speed control of machine and tool. Electric-motor drives provide suitable and quickly-adjusted speed ranges under all of the ordinary starting loads of workshop appliances. This has resulted in increasing the output from 10 to 80 per cent., compared with the old and fixed speed conditions with belts and gears.

Electric measuring instruments, such as ammeters or wattmeters, used in connection with the motors operating each machine or a small group of machines, have shown this method of transmission to result in a saving of from 20 to 30 per cent. of the power required by the old method. The electric motor is in operation and consuming energy only when needed. It is ready for instant service under any change of conditions of speed or power required by the work. It will meet easily 50 per cent. overload, and maintain it for half a day or till the whistle blows. It admits of readiest expansion of workshop facilities, and machines so driven may be placed anywhere to best advantage.

ELECTRIC, STEAM, COMPRESSED AIR, AND HYDRAULIC TRANSMISSION.

The system of electric transmission for a manufacturing establishment is the only one which admits of economically centralizing the so-called "mechanical plant" supplying light, heat and power. It is a great advantage to be able to locate the power-house near coal and water supplies. In some cases it thereby allows the use of condensing engines instead of non-condensing.

By adopting electric transmission the engine may be located at any convenient distance from the machines, while shafting transmission imperatively requires that it be located as near by as possible.

If steam power is converted into electricity for all of the mechanical operations of a manufacturing establishment, it admits of a subdivision of the generating plant into duplicate and interchangeable units, the advantages of which will be apparent. The generating plant may be operated at all times in such a manner that each engine is loaded to its normal capacity. One or more units may be held in reserve for rush work and heavy orders. Night shifts can be supplied with power in units suited to their work, rather than requiring the usual single large engine to be operated as in the ordinary day shift. Nearly uniform loads may be maintained on the engines, both day and night, the electric-light load, at night, offsetting additional electric-motor service during the day.

In some cases it may be found necessary or desirable to have a mixed system of steam and electric distribution, but it can be obviated, with increased economy, by judicious installation. The use of small steam engines about an establishment, for any purpose whatever, is to be deprecated. The maximum economy to-day is to be obtained by centralized power generation.

There are, of course, certain operations in manufacturing work best performed by compressed air, while others are best performed by hydraulic pressure. Electricity does not enter these special fields. Compressed air and hydraulic pressure provide means for performing work by the more or less direct application of the energy distributed by these fluids. They would not be seriously considered as suitable systems for general power transmission for such short distances as are required in factories and mills.

For all manufacturing operations (except the very lightest) requiring rotary motion, continuous or intermittent, uniform or variable, and reversible or otherwise, electric motors provide the readiest facilities. For certain very definite reciprocating movements, with fixed time or distance limits, hydraulic mechanisms

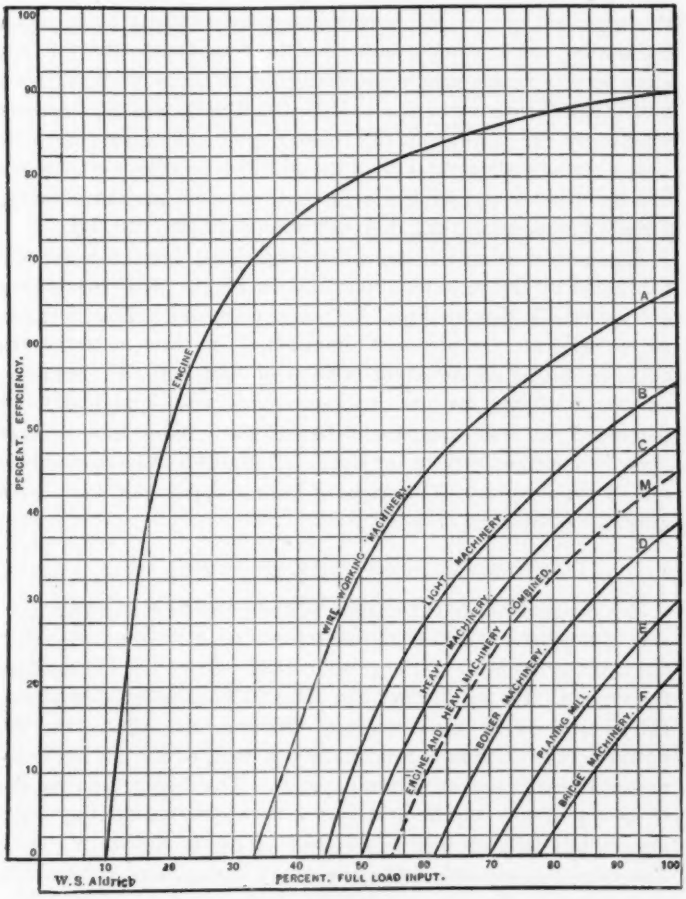


FIG. 1.

Curves showing the variations of mechanical efficiency of the engine and of the shafting transmission, at various proportional parts of the full-load indicated horse-power supplied to the system.

are best suited, though electric-motor driven hydraulic mechanisms with hydraulic control have proven admirably adapted to this class of work. For reciprocating movements requiring a cushioning effect, compressed air is best adapted. But for efficient service in any case compressed air requires to be reheated and used expansively in a cylinder with all parts and mechanisms practically the same as in the steam engine. Electric transmission is equally adapted for all of the so-called auxiliary machinery of a manufacturing establishment, the non-productive machinery, pumps, compressors, fans, blowers, cranes, hoists, lifts, etc.

The introduction of electric transmission has been greatly facilitated by the recent standardization of electrical machinery and especially that required for power purposes. This, furthermore, facilitates repairs and renewals, if it were necessary to advance this as an argument in favor of electricity. Fortunately, electric machinery as made to-day requires the simplest moving parts and the fewest repairs and renewals. Many motors have been in constant service for over six months and have required only a supply of oil. Others have been in service for years, with renewals of brushes only required.

EFFICIENCY OF TRANSMISSION AT VARIABLE LOADS.

It has usually been the custom to compare the relative efficiencies of mechanical and electrical transmission without any reference to the loads, or to the proportional part of the full load on the system. It is well known that almost all examples of mechanical friction, as in steam engines, shafting, etc., show that the power lost in friction is practically constant, within all ordinary ranges of loads and at uniform speeds. This being the case, the efficiency of any mechanical system, as an engine driving shafting, rapidly falls off below full load. This is shown in Fig. 1. The engine is assumed to be about 275 horsepower, and to have a mechanical efficiency at full load of about 90 per cent. The curves for the several classes of machinery, *ABCDEF*, show similar falling off of the mechanical efficiency of power transmission by shafting. They are plotted from average values of the percentage of total power lost in such friction for shafting trans-

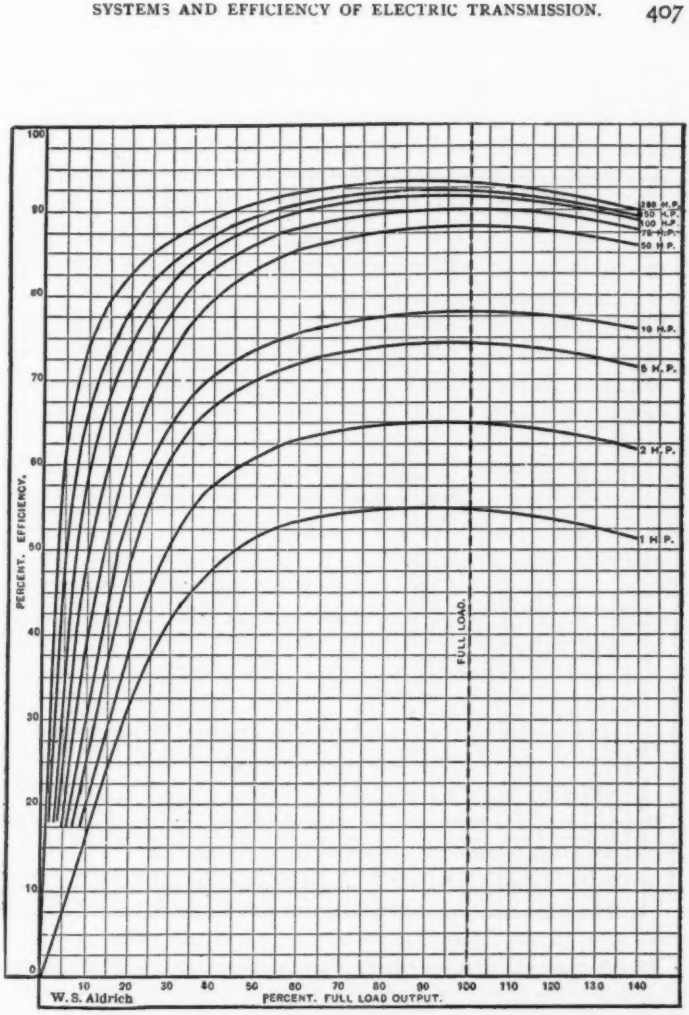


FIG. 2.

Efficiency curves of electric motors, of the same type but of different sizes, from 1 to 200 horsepower.

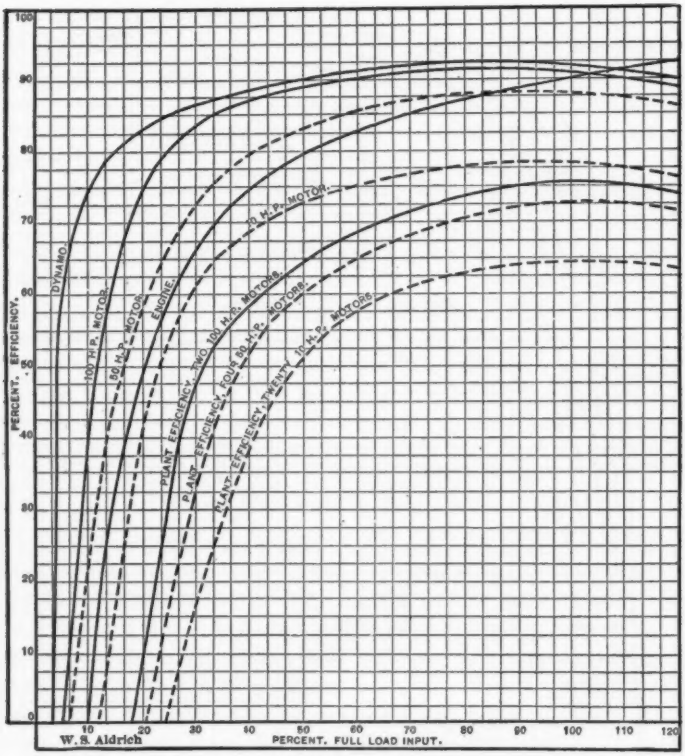


FIG. 3.

Efficiency curves of given engine and dynamo supplying power for electric motors of different sizes, with combined efficiency of electric transmission, at various proportional parts of the full load indicated horsepower supplied to the system.

missions, as given by Prof. C. H. Benjamin in a recent paper presented to the Society.* The resulting or combined efficiency of an engine driving heavy machinery, is shown in curve *M*. It is to be especially noted what little power there is left for useful work, at light loads. While at full load or 100-per cent. load there is 45 per cent. available for useful work, at three-quarters load (75 per cent.) there is only 28 per cent. available for work, and at a load of 55 per cent. of the full load there is nothing left for work, all power being consumed in overcoming the combined friction of the engine and the transmission machinery.

In the electric transmission of power for similar manufacturing purposes, the distances are so short that there is really no necessity for considering any of the so-called line losses which figure so prominently in long-distance electric transmission. The transformations of energy in the electric system from steam power into electric, thence to mechanical power, are accompanied with inherent losses, which are shown in Figs. 2 and 3. In Fig. 2 it will be noted that electric motors have the characteristic feature of high maintained efficiencies at part load. The several curves shown are for various sizes of motors of the same type of commercial machines. Of course such motors may be designed to have maximum efficiencies at lighter or heavier loads than there shown in the diagram.

Three cases of electric transmission are shown in Fig. 3, as follows: two 100-horsepower motors, four 50-horsepower motors and twenty 10-horsepower motors. The steam-engine generating plant required in each case, at normal load rating, will be of 265 indicated horsepower, 275 indicated horsepower and 315 indicated horsepower, respectively. Some interesting features of the electric system are shown in this diagram. In what is probably an extreme case, with only small 10-horsepower motors in service, and these given quite a low rating, it will be seen that at full load (100 per cent.), 63 per cent. of the power applied at the engine is available for useful work; at three-quarter (75 per cent.) load, 62 per cent. of the power is available; at half load (50 per

* *Transactions* of the American Society of Mechanical Engineers, Volume XVIII, No. 712, "Friction Horsepower in Factories," by C. H. Benjamin.

cent.), 51 per cent. of the power is available; while at somewhat less than one-quarter load all of the power applied is required to supply the losses in the engine, dynamo and motors. This may be compared with curve *M*, of Fig. 1, showing the combined efficiency of engine and shafting in the mechanical system. For 10-horsepower motors, would, in general, represent the average (normal) rating of direct-connected electric motors for such service as that designated as "heavy machinery." In fact, the 10-horsepower motor curve of Fig. 2 is quite conservative, and in many cases such an efficiency curve might easily be brought nearer the 50-horsepower curve than the 5-horsepower curve.

It is, therefore, not so much a question of what the friction horsepower (or its per cent. of the total) may be at any load, such as the normal, as it is a question of how the total or combined efficiency of the system is going to vary with the load. The best performance of the mechanical system of transmission by shafting and belts is at full load. The best performance of the electric system may be maintained from 75 per cent. to 125 per cent. of the full load—from three-quarter load to 25 per cent. overload.

These differences become more vital when it is borne in mind that very few factories and mills are operated at full load all day. For machine-shop practice, for instance, an average of about half time for the actual work would be rather liberal. In many shops several machine tools will not be worked much more than one-third of the time. The losses at these light loads are exceedingly expensive for the mechanical system of transmission, but scarcely require consideration in the electric system. In fact, by proper selection of dynamo and motors, a combined efficiency at half load may be obtained equal to that at full load. The all-day efficiency of the electric system may thus be made very high. This is quite independent of the load factor, or the ratio of the average to the maximum load during the day. The load factors may be the same for both the mechanical and the electric system, as shown by load curves, while in the mechanical system, for instance, at three-quarter load only 28 per cent. of the total power supplied will be available for useful work; in the electric system 62 per cent. will be so available, as may be seen by com-

paring curve *M*, Fig. 2, with the plant efficiency curve for twenty 10-horsepower motors, of Fig. 3.

SYSTEMS OF ELECTRIC DISTRIBUTION FOR FACTORIES.

In choosing a system of electric transmission for manufacturing work, it is not necessarily best to have that one system which will the most readily lend itself to all of the work to be performed, for light, heat and power service. A composite system may prove best suited, even in such short-distance transmission. That is, lighting service will, in general, be more satisfactory, and need not be more expensive, if supplied independently of the power service. Direct and alternating currents are equally adapted for factory transmission, and by simple or multi-circuit systems of distribution; that is, by two, three, or four-wire systems, as the case may require. Preferably, all distribution should be direct; that is, without the use of storage batteries, rotary converters, or transformers, except for certain lines of work in which it may be necessary to use one or the other of these indirect systems of distribution.

In the matter of voltages a wide range is possible: 110-volt two wire and 220-volt three-wire systems for use of either direct or alternating currents for light and power; 440-volt two-phase alternating-current three or four-wire systems for both light and power; 550-volt direct-current two-wire system, or 550-volt alternating-current three-phase three-wire system, chiefly for power service, or the monocyclic system for both light and power. In general, it will not be necessary nor advisable to use over 550 volts, direct or alternating current. Shocks arising from accidental contact with wires carrying currents of this voltage are not necessarily dangerous. Experience has shown that workmen respect the distributing wires the higher the voltage. But it is not necessary to command such respect by raising it above 550 volts.

ELECTRIC TRANSMISSION BY DIRECT CURRENTS.

At the time that electricity was introduced into manufacturing establishments the direct-current system was the only one available. For the peculiar and exacting service required in driving

all kinds of machine tools and various workshop appliances, there were difficulties to be overcome with any system. It was necessary to secure satisfactory methods of producing a large starting turning moment, or torque, for varying the speeds as might be required under uniform or variable loads, and for reducing to a minimum the trouble arising from the use of a commutator.

With direct-current motors, it was a simple matter to introduce starting boxes (resistances) in the armature circuit to control the torque, as well as rheostats (resistances) in the fields to control the speed in particular. But every such resistance meant an expenditure of energy in otherwise useless heating of the wire or other material of which these resistances might be made. The so-called Ward Leonard system came to the rescue with its two additional machines in order to operate the one given machine as a motor at practically a constant efficiency under all conditions of load and speed. This system has been very successfully and extensively used in elevators, cranes, etc. By the use of the auxiliary machines the supply voltage may be varied according to the speed desired, and the current supplied according to the torque required, without wasting any energy in heating wasteful resistances. For conditions of factory service permitting of such an application, two motors may be advantageously used on one machine or set of machines, by means of which it is possible to vary the torque and speed quite as satisfactorily as in street-car working, by the series-parallel method of control.

The difficulties with commutators have been almost entirely overcome, and many refinements in design effected, so that the direct-current motor of to-day leaves little to be desired. Such objectionable features as still remain are inherent in the direct-current system used, and are found to lie chiefly in the kind of armature, commutator and brush devices required. These parts are most liable to derangement, require systematic attention for cleanliness and efficiency and renewals of brushes.

In heavy work, with the ordinary rough usage which an electric motor receives in a factory, a commutator may become in

less than an hour so dirty as to result in considerable heating, due to increased mechanical friction of the carbon brushes, and to the increased current required by their imperfect contact. Dust-proof casings remove several of these possibilities to a more remote time. But somehow workmen expect an electric motor to take care of itself, at least they treat it on this assumption. If a motor were sure of the daily care and inspection formerly bestowed upon shafting and belting, it would have made a much better record earlier in its history. Electric motors cannot usually be similarly inspected and attended to while at work as in the case of the older system of mechanical transmission. To have such work performed at any time requires a skilled attendant.

ELECTRIC TRANSMISSION BY ALTERNATING CURRENTS. INDUCTION MOTORS.

The alternating-current system, with its induction-motor service, offered practically the only alternative to those engineers and manufacturers who did not care to be troubled with the petty annoyances and delays likely to occur at any time with the direct-current motor. The induction machine as it stands to-day is probably the most perfect motor yet developed from the standpoint of electric transmission in factories and mills. It may be started and operated from any point, at any time, at practically any load and speed within its predetermined ranges. It may be used on 110, 220, 440 or 550-volt alternating-current circuits of one, two or three phases. It does not require any direct-current supply as the synchronous motor does for its field excitation. It does not require any brushes, commutator or collecting rings. Offsetting these advantages, however, are certain restrictions. The speed of an induction motor falls off slightly as the load is increased. This is shown in Fig. 4, in which the per cent. of such change of speed is laid off at the bottom, and is known as the slip. The ability to start an induction from rest under a heavy load, as well as the possible speed changes during its operation, are obtained at some sacrifice of efficiency.

Induction motors, moreover, permit of higher lineal speeds

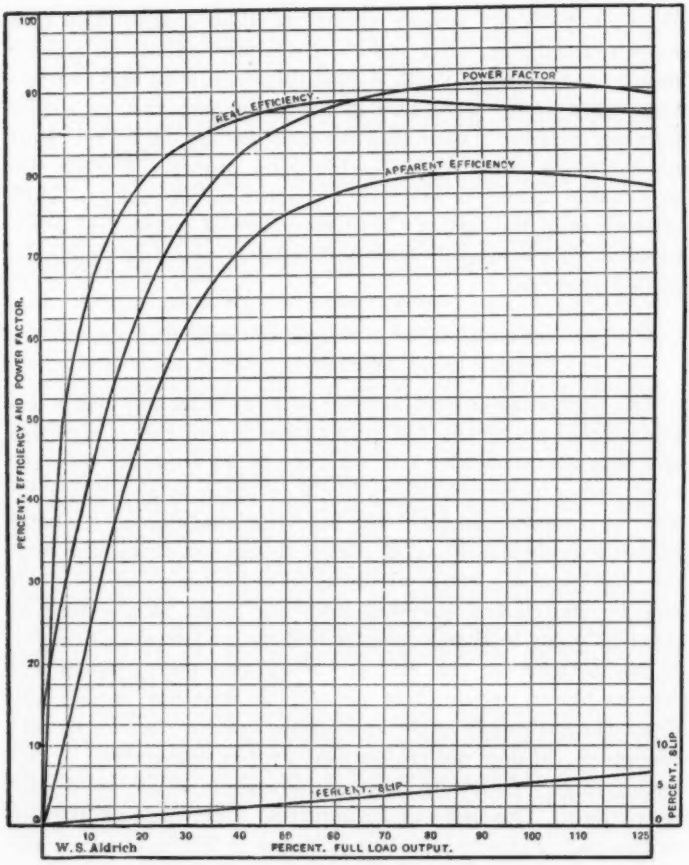


FIG. 4.

Efficiency and power factor curves of an induction motor.

than are possible with any other type, from 6,000 to 7,000 feet not being infrequent. By suitable arrangements of its field windings, this type of motor may have its speed altered in regular steps, so reducing it one-half, one-quarter, one-eighth, etc. This makes possible similar changes to gear-wheel combinations, which may therefore be eliminated to the extent that the induction motor is installed to effect such changes. In almost all cases of shop driving, the slip is not objectionable, any more than the increasing slip of the driving belt as the load is thrown on. These motors will stand almost any amount of rough usage and heavy overloads, as they cannot be burned out. If excessively overloaded, the motor slows down and stops, starting up immediately as soon as the load is lightened. Ordinarily, machine tools and almost all classes of shop machinery are started at quite light loads, and the full load is thrown on when the work or the tool is up to the speed desired. For this class of work the induction motor seems specially fitted.

A larger generating power plant is required for an installation of induction motors than would be the case if direct-current motors were used. This is on account of the energy which is lost in all classes of alternating-current circuits in which there is considerable self-induction, whether in the transmission wires or in the appliances used. In the case of induction motors this loss is very appreciable at light loads, becoming much reduced at average and heavy loads, at which it is almost uniform. It is best represented in the form of a curve of what is known as the power factor; that is, the ratio of the true energy to the apparent energy supplied to the motor. Such a curve is shown in Fig. 4, for an induction motor of about 40 horsepower. The apparent efficiency is the ratio of the output, or delivered horsepower to the apparent energy supplied to the motor, while the true or real efficiency is the ratio of the same delivered horsepower to the true energy supplied to the motor. Curves are also shown for these values, in Fig. 4, for the same motor.

SYNCHRONOUS MOTORS.

Synchronous motors are admirably adapted to factory service where absolute uniformity of speed is required, and where the

extra installation of a direct-current supply for their field excitation is not deemed objectionable. While induction motors are always wasteful of some energy, through their high self-induction, synchronous motors may on the other hand be brought into that condition of operation practically equivalent to the use of direct-current motors, at least for a large range of their loads. In other words, the power factor of a synchronous motor may be made almost anything from zero to unity, according to the extent of excitation of its fields by the direct current applied for this purpose.

When made in the revolving field type, synchronous motors are self-starting from rest at light loads. They may be very heavily overloaded, without falling out of synchronism or out of step, and when they do for an instant, they may be brought back again by throwing off some of the load. A well-designed synchronous motor will carry at least three times its full normal load, and not drop out of step. If an induction motor is built for such overloads it is likely to have quite low efficiency at ordinary loads.

The synchronous motor curves in Fig. 5, for a machine of about 40 horsepower, show several of the characteristics which have been previously noted. It is assumed that the motor has the same real efficiency in the two cases of different power factors, shown by their respective curves P and Q. The more nearly the power factor approaches unity, the more nearly the conditions are realized which are found in direct-current motors. The generating plant may thus be much reduced in size, compared to that required for motors with low-power factors. In fact, the power plant need not be larger than ordinarily installed for direct-current working, where the synchronous motors are properly designed, installed and operated under the best conditions.

Higher efficiencies may be obtained with synchronous motors than with induction motors of the same output. In fact, such motors realize the ideal conditions of motor working in which the motor attains almost the same efficiency as the generators. Both induction and synchronous motors have usually higher

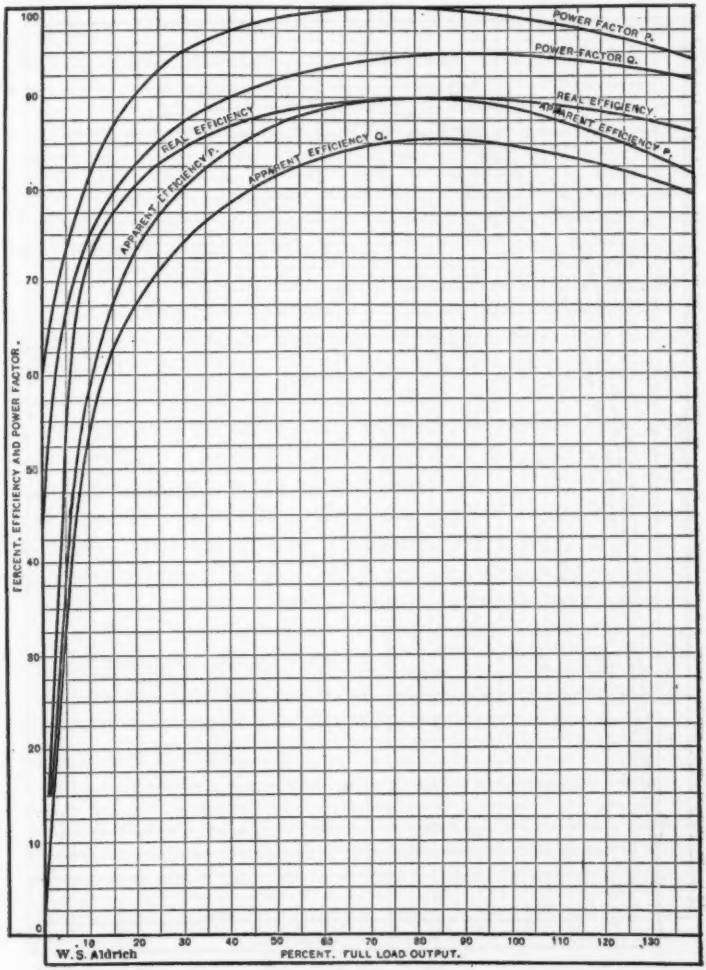


FIG. 5.

Real or true efficiency curve of a synchronous motor for two different power factors, and apparent efficiencies.

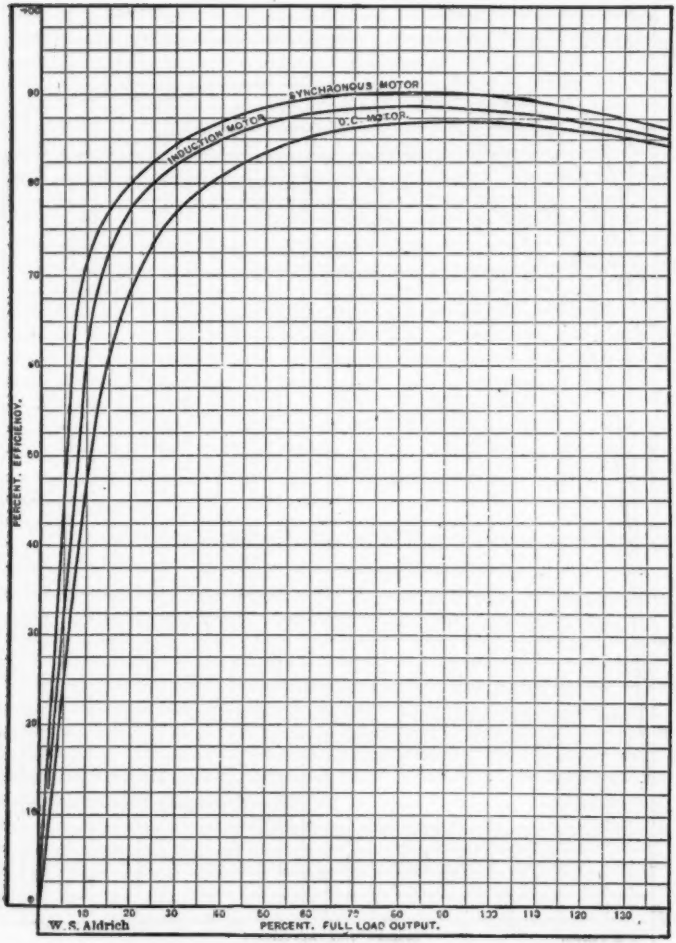


FIG. 6.

Comparative curves of efficiencies of synchronous, induction, and direct-current motors of the same size.

efficiences than direct-current motors of same size, as is shown in Fig. 6. These three curves represent the efficiencies plotted for the 40-horsepower motor, in Figs. 2, 4 and 5, respectively.

COMBINED INDUCTION AND SYNCHRONOUS-MOTOR WORKING.

The ideal conditions in a factory installation no doubt would be secured where both induction and synchronous motors were used, the former for small machines and direct driving, the latter for operating a set or group of machines. The synchronous motors would be started up just before beginning the work of the day, have at all times a light constant load, and might easily be so regulated as to produce an almost balanced system in combination with the induction motors. In such a system of transmission the lagging currents of the induction motors would be offset by the leading currents of the synchronous motors, if the latter were operated to produce such leading currents. The whole system would be operated practically throughout quite a range of load variations, as if it were a simple direct-current system. The advantage of such a condition is apparent: it means least installation for any given output, or greatest output for any given capacity of generating plant. The group method of electric driving is much better adapted for small machines, up to and including 2-horsepower capacity, and especially where such machines are in almost constant service. Above this size, individual motor driving becomes more and more efficient, particularly if the machines are operated only a fraction of the day.

ECONOMIES EFFECTED BY ELECTRIC TRANSMISSION.

It might naturally be expected that the greatest saving would be noticed in those factories designated in the earlier part of the paper as engaged in heavy machine work, where the loss in shafting and belting amounts to over 50 per cent. of the total power developed at the engine. On the other hand, many small industries have introduced electric driving to marked advantage. A few cases may be cited. No doubt many similar cases may be added by members of the Society from their own experience with electric driving.

Case I.—Electric driving reduced the cost one-half over the former method, for engineer, coal and water.

Case II.—Fuel account, by electric driving, reduced to one-fifth of what it formerly was by shafting and belting.

Case III.—Cost of power 44 per cent. lower than what it was by shafts and belts.

Case IV.—Gross saving about 30 per cent. with direct-coupled motors, and 22 per cent. with geared or belted motors.

HOT-WATER HEATING FROM A CENTRAL STATION.*

By H. T. Yaryan, Toledo, O.

It does not require the eye of a prophet to foresee the future means of heating and lighting our cities. If heat, using water as a vehicle, can be distributed from a central station covering a territory measured by a radius of one and a half miles from the station, and do it with profit to the investors and entire satisfaction to the consumers, the problem would seem to have been solved. This has been done by the Toledo Heating and Lighting Company for the past five years, a period sufficiently long to place the enterprize beyond the experimental stage, and entitles it to be classed as a successful business venture. It is because I believe the members of this Society will in the future be called upon to exercise their skill and knowledge in refining and perfecting the system that I lay the facts and "state of the art" before you, giving freely results of my own experience, in the hope that it will be of assistance when you are called upon for advice.

I think it will be found that when the new system is generally introduced, in connection with the production of electricity, you will be forced to modify your views on high-pressure steam and the compounding of engines. Where steam economies are of little avail seven months out of the twelve, and where seven months cover four-fifths of the electric output for the year, there would seem to be little encouragement for heavy outlay in that direction. Fortunately for the system, the demands for heat and electricity run almost on parallel lines a large portion of the year. Under the new system electricity becomes a by-pro-

* Paper presented at the Cincinnati meeting, May, 1900, of the American Society of Mechanical Engineers.

duct, the only cost of fuel being the condensation of engines, which will average probably 20 per cent. Experience has shown that for every dwelling house heated it requires the exhaust from engines necessary to light four. It therefore follows that the lighting district must be extended beyond the heating, or some other use be found for the output, such as street railroads, if the entire city is to be heated. If some great genius, like an Edison or a Tesla, were to announce to this Society that electricity could be produced without cost, I imagine there would be a sensation; and yet the Toledo Heating and Lighting Company is doing this very thing, as the six months' business ending January 1st last shows, the load being 199 houses heated and 450 lighted.

Receipts from heating		$9,900.88
Receipts from lighting		9,064.78
Fuel	$5,040.63	
Wages and salary	2,827.14	
Expense (taxes, repairs, oil, waste, etc.)	1,896.77	
	$9,764.54	

I presume the question will be asked why electricity should be treated as a by-product instead of the heat. My answer is, that charges for heat can be fixed and maintained, the consumption of coal increasing proportionately with the number of consumers, while the opposite is true of electricity, and a flexible scale of charges is necessary if the large excess of output is to be disposed of.

Most of you will recollect the disastrous failure in Boston, about twelve years ago, in an attempt to distribute hot water from a central station. A large sum was invested, and the work of installation was of the highest order. I have no direct information as to the causes of this failure, but it is not hard to figure out if one reads the paper of Chief Engineer Abbot before the Boston Society of Civil Engineers. The two mains laid were 4 inches for outflow of hot water, and 6 inches for return. The proposition was to heat water to a temperature of 400 degrees Fahrenheit, and to circulate, necessarily, under a pressure of 250 pounds. He says in his paper, that the maximum rate of circulation would

be from 10 to 15 feet per second. If I can figure friction, and making a liberal allowance for the less viscosity of hot water, there would be lost in pressure from 3½ to 8 pounds per 100 feet, or at the end of a pipe 3,000 feet from station from 100 to 240 pounds. The effect of this on superheated water would be to convert a large portion of it into steam, and the small service pipes of ¼ to ⅜ inch to the various houses would not deliver a sufficiency of the mixed steam and water to heat the buildings. Other troubles would follow in the return pipe. The theory was, that when the pressure was reduced inside the buildings, the superheated water would give up a large per cent. of steam, and this was to be used for heating; the water returning to the station by gravity, in a closed circuit, to be again superheated. Unfortunately superheated water will give up its steam inversely as the pressure, and as the pressure must be greater at the extreme end of the line than at the station, it follows that, as the return flow reaches points of lower pressure, steam will again be formed until the pumps reach their capacity of handling the mixed volume of steam and water, when circulation would slow down and practically cease. There are many other troubles which they must have encountered that I could mention, but the one given is sufficient, as it was inherent and fatal to the working of such a system. The surprise is, not that the experiment was a failure, but that intelligent engineers should have given their sanction to such an undertaking.

The system I am about to describe . . . . is so simple that but few words are necessary to make it intelligible. The steam boilers, engines and dynamos are such as may be used in the ordinary electric-light station. Heaters of the tubular type, through which the water passes from the pumps to the mains, receive the exhaust steam from the engines, heating the water to any desired temperature. When more exhaust is being produced than is required to heat the water, the excess is delivered to a water-storage tank to be used later when the output of electricity is small. The circulating system consists of two wrought-iron pipes, laid side by side in the ground, carefully protected by insulation, one pipe for the outflow of hot

water impelled by the pumps, the other for the return water from the coils in the various houses heated, going back to the suction end of the pumps, to be forced again through the heaters, where the loss in temperature is restored.

The Toledo Heating and Lighting Company has two stations, known as the Floyd Street and the Twenty-second Street Stations, and although the heating mains of the two stations are separate and distinct, the dynamos of the two stations feed into a common system of electric mains. The length of the double water mains, that is ditch, connected to the Floyd Street Station is 8,550 feet, and the number of houses heated is 101. The length of mains from Twenty-second Street Station is 8,260 feet, and the number of houses heated is 98. The amount of radiating surface connected with the Floyd Street Station is about 90,000 square feet, and with the Twenty-second Street Station about 85,000 square feet. The amount of coal consumed at both stations during the heating season, from October 1st to May 1st, averages 5,000 short tons of coarse slack, or about 25 tons to the residence. These same buildings, if heated by their own furnaces, would average not more than 18 tons of hard coal each. The difference may be accounted for by the loss of heat in ground radiation, and largely by the fact that we heat the buildings more thoroughly and for a greater number of hours than they could or would do by private furnaces. The loss of heat through radiation in our mains is an important one, and I am sorry not to be able to give positive data, but it is impossible to more than approximate the loss. From careful observation, I estimate the loss to be between 15 and 20 per cent. The water reaches the extreme end of our lines, three-quarters of a mile from the station, in the coldest weather, with a loss of 12 degrees Fahrenheit, which would be an average of 6 degrees to all of the houses. As the water returns to the station with a drop of 35 degrees, this would indicate about 17 per cent. loss in the ground, which I believe to be nearly correct. A pressure of 60 pounds is maintained on the feed line during cold weather and 40 pounds during moderate weather. The service pipes to the various houses are 1-inch pipe, and the return line throttled with

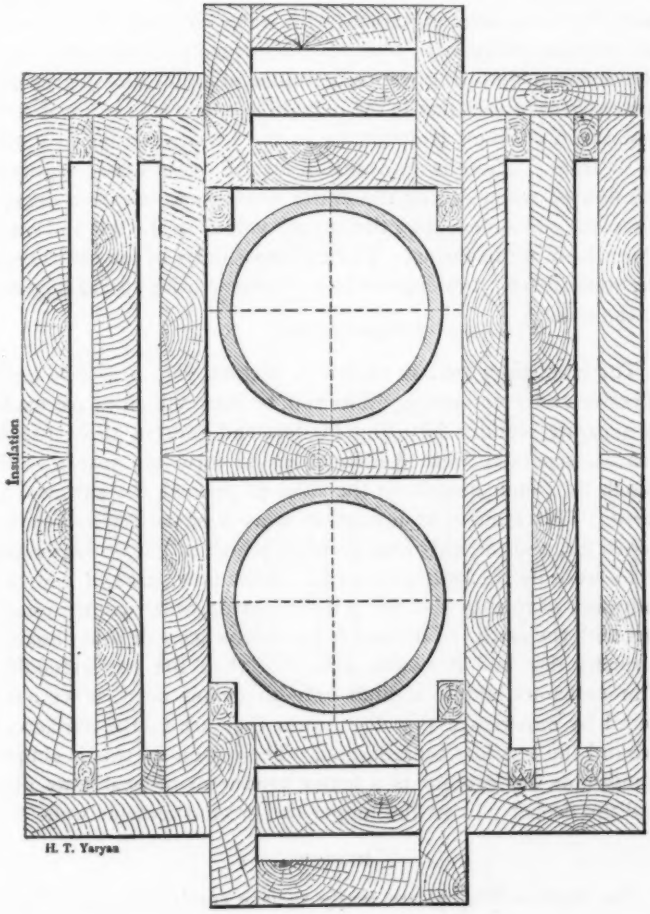


FIG. 4.

a disc inside the building, the size of opening depending upon the quantity of radiation, but average $\frac{3}{8}$ of an inch. The houses are equipped with radiation sufficient to heat them to a temperature of 70 degrees Fahrenheit, with water entering the house at 160 degrees Fahrenheit, when the outside temperature is freezing. By raising or lowering the temperature of water one degree for each degree of variation in the outside temperature, we are able to maintain a constant temperature in the houses during all kinds of weather. There is no occasion for the consumer to regulate the flow of water during the entire winter, and with few exceptions our consumers open the supply valves in the fall and only close them in the spring. The extreme limits of temperature of the water is from 130 degrees in moderate weather to 212 degrees in the coldest.

INSULATION.

The insulation used is shown in section, Fig. 4, and is both effective and inexpensive. The top, bottom and sides are made in separate sections by using for top and bottom three 1-inch boards, separated with $\frac{1}{2}$-inch strips to form air spaces, with 1-inch by 4-inch boards on the sides to prevent the entrance of earth. This manner of insulation is as near perfect as can be hoped for, and the only improvement which might be suggested is to creosote the outside boards. After four years of use, the only sign of rotting is in the outside boards which come in contact with the earth. The heat in the mains keep all other boards perfectly dry, and it would seem that their life was indefinite. The mains are laid in a ditch only 30 inches deep, as we have found by experience that the nearer they are to the surface of ground the less loss by radiation. The reason for this is simple enough, as wet ground is a better conductor of heat than dry, and the deeper you go the longer the ground remains wet.

HEATERS.

The heaters (Fig. 5) are really surface condensers, and I have some hundreds in use for this purpose, the only novelty about them being the manner of providing for the difference in expansion of copper tubes and iron shell. No thimbles or other

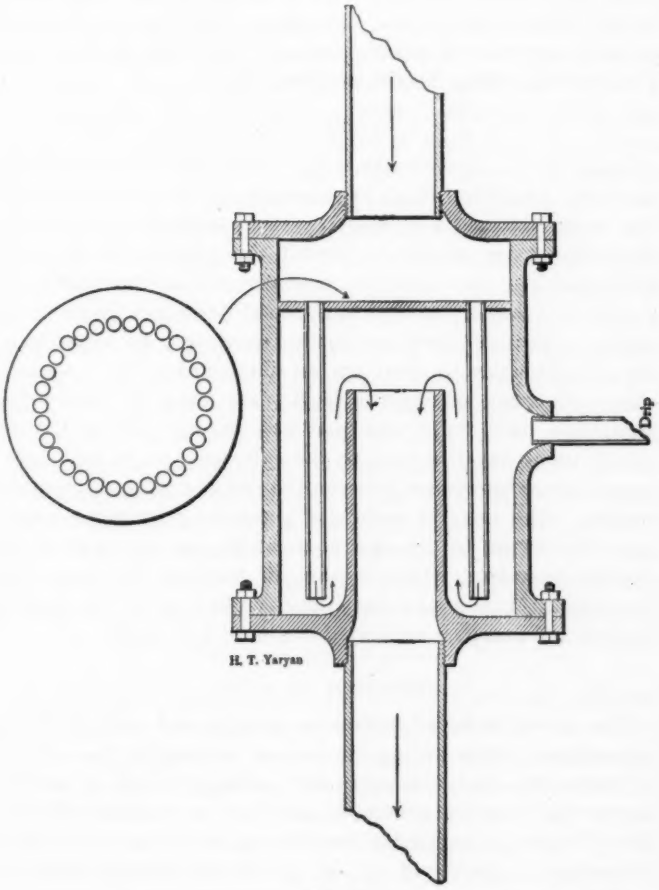


FIG. 5.

devices are used to permit a slip through flue sheets, but each tube is rigidly expanded in the top and bottom flue sheet. They are 7 feet long, 24 inches in diameter, and contain 500 $\frac{5}{8}$-inch copper tubes. The tube before entering is bent so as to make a permanent sag of $\frac{5}{8}$ inch, which is intensified to about $\frac{3}{4}$ inch in expanding, by the use of a drift.

GREASE TRAPS.

As all the water of condensation is sent back to the boilers, a thorough separation of oil is necessary. I have found most of the so-called traps a delusion, as they depend upon slowing down the steam current to permit the particles of oil to be dropped. The finer particles, however, are so thoroughly entrained that they are carried to the condensers and thence to the boilers. The trap used was really constructed to separate entrained salt water, in distillers on shipboard under a vacuum, where the speed of vapor is high. The principle involved is separating the volume of vapor into jets by passing through 1-inch tubes equal in area to the exhaust pipe, so that only a narrow space intervenes between the end of tube and impinging surface. The smallest particle of entrained substance is carried over this space, by its own momentum, and is held by the impinging surface, where it runs off through the drain pipe. Incidentally, I may state that fully one-half of the oil used on engines and pumps is recovered and used over again.

EXPANSION ON MAINS.

The provision for expansion on underground mains is of vital importance. After trying all manner of devices, the ordinary U, where the run is straight, and 90-degree bends at corners, answer the purpose admirably and give no trouble whatever. The U's are placed about 600 feet apart, and as the extreme expansion is about $\frac{3}{4}$ of an inch to 100 feet, only $4\frac{1}{2}$ inches are necessary to be taken care of by each U. As the mains are laid cold, we put an extra strain when laying on the U's in an opposite direction to that in which they work when allowing for expansion.

CORROSION OF MAINS.

Almost the first question asked by a practical engineer is, How about the corrosion of your pipes? I am happy to say that I can answer this question in a satisfactory manner, as over 10,000 feet of pipe were relaid last summer, after being in use five years. The wrought-iron pipe showed no signs whatever of corrosion, but steel pipe was perceptibly affected. There was a great difference in different lengths of steel pipe, but all were affected more or less. The appearance of the pipe showed the formation of nodules on the inside, varying in number from one dozen to one hundred in a length of pipe, and in size from a grain of corn to a hazel nut. When the nodules were removed, a soft underlayer of carbon remained, and with the finger nail this could be removed, developing a perfectly round pit from $\frac{1}{8}$ to $\frac{3}{16}$ of an inch deep, but in no instance had the pit perforated the pipe. As a matter of curiosity, some steel pipe which had been in use two years was examined and found to be in exactly the same condition as that of five years' use. The action seems to be electrolitic, some particle of impurity forming the nucleus for decomposition of the steel. If any inference can be drawn from the facts stated, it is that the impurity has lost its power to cause electrolysis after being surrounded by a sufficient amount of graphite. The most important lesson taught, however, is that only wrought pipes should be used for water circulation.

FRICTION LOSS IN MAINS.

A test with pressure gage on the outflow pipe at the extreme end of our lines shows 39 pounds with 60 pounds at the station, or a loss of 21 pounds due to friction. At the same time the pressure on the return line was 28 pounds, and as the friction should be the same on both lines, the difference may be accounted for by the fact that the return line discharges into a tank 15 feet high at the station.

In calculating the size of mains for a given number of houses, it became necessary to know the relative friction of hot water as compared with cold. To my surprise, I was unable to find any data that were of value, and to settle the question I made some

tests which may be useful in future work. A run of 450 feet of 1-inch wrought-iron pipe with numerous bends was connected to a heater, and the water passed through a meter. With a constant pressure of 10 pounds the following cubic feet of water were delivered at the open end of the line in five minutes.

At 34 degrees Fahrenheit..... 6.2	At 150 degrees Fahrenheit..... 6.6
At 60 degrees Fahrenheit..... 6.3	At 185 degrees Fahrenheit..... 6.75
At 90 degrees Fahrenheit..... 6.4	At 212 degrees Fahrenheit..... 7.9
At 120 degrees Fahrenheit..... 6.5	

As it requires an average of 15 gallons per minute to be delivered at each house, and knowing the friction of cold water, the data thus obtained enabled me to calculate the size of mains necessary to give good service to each house, and to always maintain a sufficient difference in pressure between feed and return mains to insure rapid circulation.

STEAM *VS.* HOT WATER.

As exhaust steam heating from central stations has been introduced to a limited extent, it may not be out of place to call attention to the objections to steam which do not apply to hot water.

1. The exhaust not being uniform in power plants throughout the entire period of 24 hours, either live steam must be used during the minimum load or exhaust must be lost during the maximum load.

2. The excessive cost of laying steam mains, which must be large and thoroughly protected from infiltration of water. The temperature of pipes being above the boiling point of water, any infiltration is converted into steam, and thus a vast amount of heat is dissipated.

3. It is absolutely necessary that from 5 to 15 pounds back pressure shall be carried on the engine, which is exceedingly wasteful.

4. The condensation of steam is wasted, and with it a large percentage of heat.

5. The piping of houses is expensive on account of the large

size required to convey steam with 1 or 2 pounds back pressure.

6. The service is not satisfactory on account of difficulty in regulation, especially in mild weather.

7. The offensive smell, due to vents and leaky stuffing boxes.

8. Water hammer, due to condensation in house pipes.

9. The quality of heat, especially where direct radiation is used.

10. Danger from fire, which has been abundantly demonstrated, can arise from long continued contact of steam pipes with wood in an enclosed place.

11. Greater loss by radiation in ground, due to an average higher temperature of mains.

12. The necessity for sewer connections to remove condensation in mains where traps may form.

13. The rapid destruction of all iron drips, due to the solvent action of distilled water.

14. The filling of the entire system with air when a shut down occurs for any length of time.

The only counter charge I have ever heard made was, that it required live steam to run the pumps. But this is largely offset by the back pressure on engines in the steam system and can be made a large source of profit by selling the heat, as none is wasted.

During the past two years the system I have described has been installed, and is now in successful operation in the following cities and towns: La Porte, Ind.; Matton, Ill.; La Crosse, Wis.; Kenosha, Wis.; Alton, Ill.; Portage, Wis.; Boone, Perry, Ida Grove, Iowa Falls, and Mason City, Iowa.

During the present summer the company which I represent will install a refrigerating plant for domestic use, distributing cold brine in exactly the same manner that we do hot water, but using a separate set of mains. We shall use an ammonia-absorption machine, operated with exhaust steam, so that as the demand for heat ceases that for cooling purposes begins, thus utilizing the exhaust steam from our engines during the summer months, which now goes to waste.

NOTES.

MAY MEETING OF THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS.

The forty-first meeting of the American Society of Mechanical Engineers was held at the Grand Hotel, Cincinnati, O., May 15 to May 18, 1900. The programme included twelve papers and a topical discussion on "What Does It Cost to Run Trains at High Speed." Among the papers were the following: Robt. H. Thurston, "Reheaters in Multiple-Cylinder Engines;" Geo. I. Rockwood, "On the Value of a Horsepower;" H. T. Yaryan, "Hot Water Heating from a Central Station;" W. S. Aldrich, "Systems of Efficiency of Electric Transmission in Factories and Mills;" Arthur Herschmann, "The Automobile Wagon for Heavy Duty;" B. C. Ball, "Cylinder Proportions for Compound and Triple-Expansion Engines." A number of excursions were made to points of interest in Cincinnati and vicinity.

We have included, as reprints, three of these papers in our present number, selected with a reference to their brevity as well as to their importance, owing to limited space available. We are, however, unable to append the discussion upon any of them at this time.

TIME AND COAL CONSUMPTION FOR THE VOYAGE OF THE *OREGON*, MARCH TO MAY, 1898.

A report was recently called forth relative to the cost of coal consumed by the *Oregon* in her famous trip from San Francisco to Key West, Fla., during the Spanish war. A comparison was drawn between this and the estimated cost of fuel had it been possible to have made the trip via a Nicaragua or Panama Canal. The required time for the voyages was, of course, likewise compared, the latter point being by far the most important in war times, and the following tabulated data will be found full of interest:

U. S. Battleship *Oregon*. Trip from San Francisco to Key West. March 19 to May 26, 1898.

From—	Trip to—	Date, 1898.	Knots run by log.	Coal used.			Knots run per ton of coal at sea.	Coal received at starting point Tons.	Cost of coal.	
				At sea.	In port.	Total.			Dollars per ton.	Total received.
San Francisco	Callao	March 19 to April 4	4,099.9	989.8	...	989.6	4.143	1,567.4	7.09	$11,112.65
Callao, Peru	Sandy Point, Chili	April 7 to 17	2,675.7	870.9	42.0	912.9	3.072	1,136.0	15.02	17,076.20
Sandy Point, Chili	Rio Janeiro	April 20 to 30	2,247.7	681.0	48.4	729.4	3.300	490.0	21.89	10,730.63
Rio Janeiro	Barbadoes, W. I	May 4 to 18	3,045.9	882.0	82.2	964.2	3.453	1,020.0	9.75	9,945.00
Barbadoes	Key West	May 18 to 26	1,886.1	579.1	27.4	606.5	3.257	241.0	13.22	3,187.20
Total			13,955.3	4,002.6	200.0	4,202.6	...	4,454.4	...	52,051.68
Coal on hand at arrival in Key West							...	251.8	7.09	1,785.26
Coal used for trip							...	4,202.6	...	50,266.42
Average price of coal used per ton							...	...	11.96	.....
Average knots per ton at sea							3.487	...	...	.....
Cost of coal per knot at sea							...	...	3.45	.....
Cost of coal per knot (including stoppage)							...	...	3.60	.....

ESTIMATED COAL CONSUMPTION FOR A VOYAGE OF THE U. S. S. *OREGON* FROM SAN FRANCISCO, CAL., TO KEY WEST VIA A NICARAGUA OR PANAMA CANAL.

	Nicaraguan Canal.			Panama Canal.		
	Distance. Miles.	Coal. Tons.	Coal cost.	Distance. Miles.	Coal. Tons.	Coal cost.
San Francisco to Western end of canal.....	2,700	651.7	$4,620.11	3,277	791.0	$5,608.19
Through canal.....	170	41.0	290.69	50	12.1	85.79
From Eastern end of canal to Key West...	965	232.9	1,651.26	1,068	257.8	1,827.80
Total.....	3,835	925.6	6,562.16	4,395	1,060.9	7,521.78
Delays at 15 tons per day	.....	45.	319.05	.....	30.	212.70
Total.....	.....	970.6	6,881.21	.....	1,090.9	7,833.48
Coal remaining on board at end of trip..	.....	596.8	.....	.....	476.5	.....
Saving in cost of fuel by going through the canal	.....	.....	43,385	.....	.....	42,433

COMPARISON OF TIME OF VOYAGES.

Actual time taken to make the trip through the straits, days.....	68
Average speed with coal taken on board in San Francisco, per day, knots.....	260
Time by Nicaraguan Canal (including delay of three days getting through Canal), days.....	17
Time by Panama Canal (including two days delay getting through Canal) days..	19
Saving in time by Nicaraguan Canal, days.....	51
Saving in time by Panama Canal, days.....	49

It should be noted that the *Oregon* could carry more than an ample supply of coal for the entire voyage via either canal route and thus secure the benefit of the cheaper price of coal per ton existing in home ports.

Another point not here considered, is that of the possibility of excessively high canal dues being current during war times, unless our own government should have control of the canal.

THE IMPERIAL JAPANESE NAVY.

In a recent paper by Rear Admiral C. C. P. FitzGerald, R. N., read before the Institute of Naval Architects, England, the rapid

growth of the Japanese navy is followed from the date of the first steamship possessed by Japan up to the present year of excellence of that nation as a naval power. We quote from "Engineering" the following extracts from this interesting review:

"In July, 1858, Her Majesty Queen Victoria presented the Mikado with a small steam yacht of about 400 tons. She was called the *Emperor*, and was presented at Yeddo (Tokio) by Lord Elgin to the Japanese Commissioners deputed by the Shogune to receive her. The Mikado did not make use of her; at that time he was not allowed out. I believe this to have been the first steamship possessed by Japan.

"The first real start made by Japan in the production of a modern Navy seems to have been the purchase of the ironclad *Stonewall Jackson* from the United States Government in 1866. She was a small ship of only 1,300 tons burden, but she carried a 10-ton gun, besides some smaller ones, and was a powerful ship of her day; she was renamed the *Adzuma*.

"The first ship built in England for the Japanese Government was the *Foo-So*. She was built at Poplar by Samuda, from designs by Sir Edward Reed, and was launched in April, 1877. She was a broadside, central-battery ship, bark-rigged, 220 feet long, 48 feet beam, 3,718 tons, double screw, speed 13 knots, engines by Penn. This ship was followed by the *Kon-go*, *Hi-yei* and *Rin-jo*, all small ironclads not exceeding 2,300 tons, but carrying powerful armaments for their size. There were also about half a dozen unarmored ships of little fighting value. This was the state of the Japanese Navy in 1880. Five years later (1885) Japan had only added one small ironclad to this list; but there were "built and building" for her several fast and powerful cruisers, armed with Krupp and Armstrong guns. The ironclads, with the exception of the *Foo-So*, were built of wood. Five years later (1890) she had again only added one ironclad to her list, in the shape of an armored gunboat; but she had by this time provided herself with a considerable squadron of fast and well-armed cruisers, built in various foreign countries.

"By 1895, although she had not actually added to her list of armored ships, there were building for her in England two bat-

tleships of the most powerful type, exceeding 12,000 tons displacement, and with a proposed speed of 18 knots; she had also added considerably to her list of fast cruisers. One of these—the *Yoshino*—built at Elswick, had a measured-mile speed of 22.5 knots. There can be no doubt that the Chino-Japanese war gave an immense impetus to the development of the Japanese Navy; not only were ships captured from the Chinese, some of which were repaired and are now in commission, but large orders were placed abroad for warships of all classes, including torpedo craft, and the Japanese also set to work to build ships in their own dockyards. The Japanese Navy now stands as follows, eliminating ships which appear to be of insignificant fighting value, but including those which are expected to be ready during the current year:

Battleships.

Fuji, ...	Thames Iron Works.
Yashima, ...	Elswick.
Shikishima, ...	Thames Iron Works.
Asahi, ...	John Brown & Co.
Hatsusi, ...	Elswick.
Mikâsa, ...	Vickers & Co.

"These are first-class battleships in the fullest sense of the term, ranging in tonnage from 12,300 of the *Yashima* to the 15,000 of the *Asahi*, *Hatsusi* and *Mikâsa*. The speeds are all at least 18 knots, and they are armed with the most powerful modern guns, and considerable areas of their sides are protected by the latest and most up-to-date face-hardened armor. Four of the six carry more armor and more guns than British first-class battleships, but less coal. There is also the *Chinyen* (late *Chin Yuen*), captured from the Chinese, German built. I have seen her quite lately. She has been thoroughly repaired, and is now in commission; and, although she cannot be classed as a first-class battleship, being of only 7,220 tons, and 14 knots speed, she is a powerful ship of her class. There are also three small ironclads—*Fuso*, *Hi-yei* and *Kon-go*—built in England in the seventies (before alluded to), and the *Hei-Yen*, captured from the Chinese; they

are of very small fighting value, and three of them are used as training ships.

"*Armored Cruisers.*—Although Japan won the battle of Yalu with second-class cruisers fighting against armored ships, her statesmen are not under the delusion that second-class cruisers will be sufficient to meet the growing needs of their rapidly expanding empire, and they are, therefore, adding to the fleet six very powerful armored cruisers of about 9,800 tons displacement and about 20 knots speed:

	Tons.	Speed.	Armament.
Tokiwa, ...	9,750	21.5	Four 8-inch, fourteen 6-inch, and twelve 12-pounders.
Asama, ...			
Idzumo, ...			
Iwate, ...			

These are all Elswick ships, designed by Mr. Philip Watts. The *Tokiwa* and *Asama* are completed, the *Idzuma* will be delivered about the middle of this year, the *Iwate* towards the close of it. The *Adzuma*, of 9,436 tons, but the same armament, and 20-knot speed, is building at St. Nazaire, by the Société de la Loire, and is to be ready this year. The *Yakuma*, of 9,850 tons, and the same speed and armament as the *Adzuma*, is building at the Vulcan Works, Stettin, and I am informed she is to be ready this year.

The above ships constitute a squadron of six extremely powerful vessels, call them what you will, battleships or cruisers; at any rate, not a few of the so-called "naval experts" think such ships are fit to "lie in the line" and take their place amongst battleships. They are at least as powerful vessels as some that are classed as second-class battleships in our own and some foreign navies, and they have a great advantage in speed.

Japan owns another armored cruiser, the *Chiyoda*, built in Glasgow in 1890, with a nominal speed of 19 knots; she is a small ship of only 2,450 tons, and she cannot be assigned a very high fighting value in the present day, though she took part in the battle of Yalu.

UNARMORED CRUISERS, SECOND CLASS.

	Tons.	Speed, knots.	Armament.
Chitose, ...	4,760	22.5	Two 8-inch, ten 4.7-inch, and smaller guns.
Kasagi, ...	4,784		
Takasago, ...	4,160		

The *Chitose* was built at San Francisco, the *Kasagi* at Philadelphia, and the *Takasago* at Elswick.

Yoshino,*	4,150	22.5	Four 6-inch, eight 5-inch, and smaller guns.

This ship was built at Elswick.

Akitsushima,*	3,100	19	Four 6-inch, six 5-inch, and smaller guns.

This ship was built in Japan.

Naniwa,*	3,650	18.7	Two 10-inch, six 6-inch and smaller guns.
Takachiho, ...			

Built at Elswick in 1885.

We come now to a group of a peculiar type of cruiser, carrying one very heavy gun forward, and a battery of light guns on the main deck aft.

Matsushima,* . . .	4,210	16	One 12-inch, eleven 5-inch, and smaller guns.
Itsukushima,* . . .			
Hashidate,* . . .			

"The two former ships were built in France in 1889; the latter in Japan two years later. Such an armament appears to be out of place in a cruiser, and a nominal speed of 16 knots to be inadequate. It does not appear that this type is likely to be repeated.

"*Third-class Cruisers and Small Craft.*—Japan possesses several third-class cruisers of good speed, capable of acting as scouts. She has also a considerable number of small vessels of low speed and but little fighting value, which it would be waste of time to describe. But there are six gunboats of the *Chinto* class, captured

NOTE.—The ships marked * took part in the battle of Yalu.

from the Chinese, carrying one 11-inch gun. These might be useful as coast defenders.

"*Torpedo Flotilla.*—The peculiar nature of the Japanese coast line, with its numerous harbors, and the Inland Sea, with its archipelago of islands, are physical features in Japan which offer special advantages for the use of torpedo boats; and she is, therefore, providing herself with a powerful torpedo flotilla of the most modern type of vessels.

" Messrs. Yarrow, Thornycroft, Normand and Schichau are all building either torpedo boats or destroyers for the Japanese Government, and some are also being built in Japan. A torpedo transport on the plan of our own *Vulcan* and the French *Foudre* is projected, but my information does not enable me to state whether the order has actually been placed or not.

" Messrs. Yarrow & Co. have just completed six destroyers of 31 knots speed and upwards, and the same firm has now in course of construction ten first-class torpedo boats for the Japanese Government. Messrs. Thornycroft & Co. have also just completed six destroyers of about 50 tons less displacement than the Yarrow boats, and with speeds of 30 knots and over. Japan already has in commission, and in reserve, a considerable number of first and second-class torpedo boats, some of these being constantly used for exercise.

" It is interesting to note that an armor-plated torpedo boat named the *Katoka*, 166 feet long, with 19 feet 6 inches beam, built for the Japanese Government by Messrs. Yarrow & Co. in 1885, led the torpedo-boat attack both at Port Arthur and Wei-Hai-Wei. It seems that the Japanese not only know how to order a good article, but to use it when they get it.

"*Dockyards.*—There are three Imperial dockyards in Japan—Yokosuko, Kuré and Sassebo. They are all capable of being effectually defended. (A fourth, Maisuru, on the northwest coast of the main island, is also in course of construction.) Sassebo can only be approached through narrow and tortuous channels, and, from its natural position, may be considered absoutely unattackable from the sea. Kuré is in the Inland Sea, and its position, also, is naturally a very strong one, the islands around it are

being strongly fortified, and will shortly be impregnable to sea attack. It may, moreover, be remarked that, with the powerful torpedo flotilla Japan has already got, and is still further increasing, hostile ships operating in the Inland Sea would be likely to have a bad time. Yokosuko is in the Gulf of Yeddo, and very favorably placed for defense. The heights around are already fortified, and the works now in progress at the entrance to the gulf will protect not only Yokosuko, but also Tokio and Yokohama, and forbid this large stretch of inclosed water to any hostile squadron. Nagasaki, where there is a private shipbuilding yard that turns out large merchant steamers, and where there is one first and one second-class dock, it being strongly fortified; and from its position it is a place of strategic importance. At Hakodati, in the North Island of Japan, the harbor is being artificially improved, and, although there is no dockyard here, the port is being fortified as a harbor of refuge. This place bears a striking resemblance to Gibraltar. At Oterrani, also in the North Island, extensive harbor works are in progress, an interesting description of which was lately given in 'Engineering.'

"Up to the present the Japanese dockyards have not undertaken to build a battleship, and the largest cruiser they have built is the *Hashidate*, of 4,200 tons and 16 knots speed; but they hope soon to be able to build first-class cruisers at Yokosuko, and eventually battleships. At this dockyard there is a first-class modern dock, in which one of the heaviest battleships in the British Navy (the *Victorious*) was lately docked for cleaning purposes, and I never saw a similar operation more quickly, more quietly, nor more methodically performed in any English dockyard.

"In the 'Jiji Shimpo' (Times of Japan) of February, 1899, Mr. S. Sassow, chief director of naval construction, writes as follows concerning Yokosuko dockyard:

"This dockyard was established during the Tokugawa regency by the Shogunate in 1866. French officers, comprising naval constructors and engineers (M. Verner being the chief director) were engaged, together with a considerable number of leading workmen, for originating the work and for instructing Japanese

workmen; several wooden ships have been built here. In 1875 the services of the greater part of the French employés were dispensed with, and the administration passed entirely into our own hands. . . . We are now building entirely of steel. Our artisans in all branches of shipbuilding and engineering have now attained to a considerable skill. . . . Hitherto the limit of size to Yokosuko has been 5,000 tons; but it is intended to enlarge the dockyard so as to able to build cruisers of all classes; and in course of time we expect to be able to build battleships. All materials have to be purchased abroad, even for building cruisers."

BOILER COMPOUND QUESTION.

Mr. F. Elmo Stimpson of Chicago, president of one of the branches of the National Association of Stationary Engineers, a short time ago requested Mr. W. H. Edgar of the Dearborn Drug and Chemical Works to answer certain questions relative to the action of soda ash or caustic soda on steam boilers. Knowing that his answer was to be read at a meeting of the engineers who were giving some study to this branch of chemistry, Mr. Edgar entered into the subject very fully. He said:

"We have caustic soda, which is commonly known as concentrated lye (sodium hydrate); soda ash, which is carbonate of soda and when in crystal form is known as sal soda or washing soda; bi-carbonate of soda, which is the same as carbonate of soda except that it has only one-half the quantity of soda, the other half being displaced by hydrogen; sulphate of soda, which is commonly known as Glauber's salts; chloride of soda, which is table salt; nitrate of soda, found in large quantities in South America and used as a fertilizer, and also in obtaining or producing saltpeter; fluoride of soda, which is the combination of hydrofluoric acid and soda; phosphate of soda, which is the result of the phosphoric acid and soda; and so we have a soda base combined with these acid radicals, forming the different salts of soda. All salts of soda are soluble in water (especially in hot solutions) and do not, nor will not precipitate out of solution until the solution becomes saturated so that it will not hold a larger quantity

in solution, when the salt of soda will then cake at the bottom or hottest parts of the vessel. Sodium salts are not volatile nor will they evaporate or carry over as a gas by distillation or sublimation. So you will readily see that the soda found in the condensation in the engine cylinder and through the steam system does not volatilize and carry over or pass over by itself; it is carried over mechanically and not chemically or physically, which I will explain later on.

"In answer to your first question it may be said that if the steam is perfectly free from moisture when delivered from the boiler, it will not contain nor carry any soda, as it is the moisture that holds or contains the soda salt in solution, and if this moisture is left behind we also leave all trace of soda behind. Your second question is: 'If one pound of soda (that is caustic, sal or soda ash, or, we may say further, any salt of soda), be dissolved in a given quantity of distilled water and the water boiled and evaporated, what remains?' The answer is that your entire amount of soda would remain; there would be no loss whatever. You would recover all the salt you had in solution except as careless boiling caused the water to be spattered out of the dish; of course all solid body of moisture spattered and lost would contain just so much soda. This is the only possible way in which you could lose any part of the salt you had in solution.

"It may be stated futher that the term 'salt' means the combination of an acid radical and a base, the two forming a neutral salt. Soda combined with different acid forms sodium salts; lime combined with different acids forms lime salts; and the same with magnesia or any other base, as when combined with corresponding acid radicals they form neutral salts, named to correspond with the acid radical with which the base is combined. You understand that soda ash is washing soda, which is used to make soap. Soap is also made with caustic soda. Soap is the result of the action of the soda base combining with the fat or oleic acid, forming oleate of sodium. This gives us soluble soap. The carbonic acid gas, combined with the soda, is carbonate of soda or soda ash; or the hydrate combined with the soda as caustic soda is easily and readily displaced by the oleic

acid, the soda having a greater affinity for the oleic acid than it has for the carbonate or hydrate radical. This is called saponification. You know very well what soap would do in the boiler; you know its action in the water; it forms a suds; it foams. Now, the soda has the same action on the water without grease or soap. We do not, nor can we, get this action from a chloride, a fluoride or a nitrate of soda. These three—chloride, fluoride and nitrate—belong to the same group; their physical and other peculiarities are very similar. Combined with soda, the affinity between the soda and these acids is so great that it will not break up to go to another acid radical; consequently we cannot saponify the water and cause consequent foaming by the use of these salts nor by use of phosphate or sulphate. But a sulphate will give off to some extent, softening the water by breaking up into a hydrate when in a boiling solution.

"Now, when you use caustic soda or soda ash, introducing the same into your feed water, you saponify the water in your boiler, causing it to foam, little bubbles to form, which break up and spatter the steam, and the water being more sudsy, the steam is spattered that much more; and these little, minute, solid sheets of water are virtually blown over. The great velocity with which the steam is leaving your boiler carries this moisture over mechanically, and when you have soda in solution it is natural that any part of that solution carried over will take over just so much soda. Any grease or dirt that may be in your boiler or water will form just so much more or will cause just so much more of this saponifyable condition, causing more violence or perceptible foaming.

"Soda ash or caustic soda carried into the cylinder of the engine cuts the cylinder oil, forming soap with the animal or vegetable part of the cylinder oil, and mechanically coagulating into a sticky, non-lubricating mass the mineral part of the cylinder oil. Thus your lubrication is destroyed. The same action takes place in a boiler where you have oil coming back in the returns. Soda carbonate and soda hydrate saponify the animal oil, forming soap, which gives you violent foaming. It cuts and coagulates the mineral oil into a mass.

"A word on oils. Where you are constantly buckling the tubes in a water-tube boiler, or bagging the sheets in a tubular boiler, or where the back ends of the tubes of a tubular boiler are constantly leaking, it is due to oil coming into the boilers with the returns. The animal or vegetable oil, combining with the lime and magnesia as oleate of lime and magnesia (most of the lime and magnesia being in the form of a carbonate), the carbonic acid is readily displaced by the oleic acid. These oleates of lime and magnesia are insoluble, because the oleic acid has combined with an insoluble base. This oleate of lime is formed on top of the water in little globular masses. These little particles are very adhesive and sticky; they gather together, forming into a mass and then become of gravity equal to the gravity of the water and are carried buoyantly with the circulation. They adhere mostly around the back ends of the tubes or in the lower water tubes, because this substance has a greater affinity for the hotter metal than for the cooler parts of the boiler. Being a perfect non-conductor of heat it concentrates and checks the heat in these parts, preventing the water from absorbing the heat units. Then the metal is brought up to such a degree of heat that it melts, the pressure within forcing the metal out and buckling the tube or forming a bag. The mineral oil combines with the lime and magnesia as a mechanical mixture in a coagulated, sticky mass, incorporating with and forming a part of the general mass, all of which is insoluble and very adhesive.

"This mechanical mixture is also formed with common salt (chloride of soda), or the fluoride or the nitrate of soda. Any one of these three will form a more sticky, tenacious, adhesive, coagulated substance mechanically with the oil (both mineral and animal) and in a hot solution far worse than the combination with lime or magnesia. This is formed because the affinity between the fluorine, nitrogen and chlorine with a soda is so great that it does not break up but enters into a mass with the oil; so wherever you may be getting a little oil back in your returns, a fluoride or chloride of soda will cause your boiler to leak and bag more quickly and the action is more vicious than

the action with lime. When we say lime, we mean the natural scaling salts in your feed water. Now, where you are using the average feed water you have carbonate of lime and sulphate of lime, which form scale. And when you use soda ash it is for the purpose of furnishing carbonic acid or carbonate to give you a double salt, or enough carbonate to make your lime soluble as it originally was in solution as a bicarbonate ('bi' meaning two of the acid and one of the base). But you will readily see that you have your soda in solution, consequently you get the soda action, foaming, etc.

"Hydrate is used because lime salts are soluble in an excess of the hydrate, but this excess gives you the same trouble. Chloride of soda has no action on the lime, as the chlorine has a greater affinity for the soda than it has for lime. Fluorine is introduced with the object of converting the lime carbonate and the lime sulphate into fluoride of lime, which is flocculent light precipitate. But you will readily see that where your fluoride goes to your lime your soda goes to the carbonate with which the lime was combined and leaves the soda ash in solution; so that at the end of two weeks or one month's run your solution in the boiler is saturated with soda ash. Then you get a soda action, your water is saponified and carries over, your valves and connections are all eaten out from the same old soda action. Then again the fluoride of lime, which is formed and which would remain as such in a cool solution, is taken up to a great extent in a hot solution, due to high heat in a steam boiler, which the average chemist does not stop to figure and think of. When this fluoride of lime breaks up you get the oxide of lime and free hydro-fluoric acid. The hydrogen of the water combines with the fluorine, forming hydro-fluoric acid; and the oxygen of the water goes to the lime, forming oxide of lime. This hydro-fluoric acid liberated, attacks the metal and eats the iron, it being much more corrosive and vicious than even sulphuric acid. The oxide of lime forms a scale just as readily as the carbonate.

"Soda is so extremely cheap that the fakirs can afford to sell any preparation of soda and make a thousand per cent. profit;

and the prices that they get are such that all they need to do is to sell one or two kegs per week. They care not for your boilers, and if they did their knowledge would not help them out. They know nothing about the article they are selling, but they talk in the same strain that has attended the sale of all soda dopes for the last twenty or thirty years. If you are going to use soda, buy your own. Phosphate of soda, whether it be tri-sodium phosphate or the other soda phosphates, is used with the same purpose in view, namely, that of converting the lime carbonate and the lime sulphate into phosphate of lime, known as or claimed to be tri-calcium phosphate. But you will readily see that the phosphate going to the lime, the soda must go to the carbonate, sulphate or hydrate; and you get the same action with the same results, the soda ash being left in solution. You will notice that they all explain to you the action on the lime, but they do not tell you what becomes of the soda. The man who is selling it does not know. The chemist who got it up does know, but he takes good care that you do not know through any fault of his.

"When a water is too pure and does not contain lime or some proper substance (such as the return condensation or distilled water), it attacks the iron. The action is termed electrolysis—more commonly speaking, galvanic action. When you add soda to the too-pure water—enough soda so that you have an excess of same over the lime—you then have a water too pure, an unsatisfied water, which immediately acts as the electrolyte between your poles, giving you the battery connected, and your positive pole, which is always the iron, is destroyed. Then you have pitting, grooving, etc., and general action on the joints and connections takes place. The only salt of soda that can be successfully used is the tannate of soda, and then only when used with four times its weight of tannin; also with a large excess in weight of sugars, mixing the sugars to handle the sulphates and the tannings to handle the carbonates. These mixtures can be used without any foaming or deleterious action to the iron.

"This is a subject in organic chemistry all by itself. You can buy these substances at the wholesale drug store. The reason

these soda dopes and soda salts have been used so extensively throughout the country, in the face of the fact that the joints are all leaking and you have the steam connections clamped up with leak protectors, is the fact that the consumer has been under the persuasive power of salesmen whose business is to talk nonsense and keep the engineer worked up on alleged theory. Engineers should read chemistry; figure out the reactions, and remember that there is always a by-product and a main product, and when the main product is figured out we must be sure to figure what becomes of the by-product. When we take we must give in all chemical reactions. We must take and if we do not give we must produce."—"Marine Review."

LIQUID AIR AS A MEANS FOR THE MANUFACTURE OF OXYGEN.

Readers of the "Indicator" may remember that in the numbers for April and July of 1899, beginning on pages 113 and 239, there were published two articles by the present writer in which the extravagant claims which had been brought before the public for "Liquid Air" as a source of energy, were discussed and disposed of.

Since that time, until recently, those interested in the promotion of companies which propose to establish a business on the manufacture and use of liquid air as a foundation, have not been prominently before the public.

Quite recently, however, extensive advertisements and circulars have brought the subject into notice along new lines.

Though the language of these is largely indefinite or ambiguous, yet it does not seem that the former fallacies have been definitely repeated, though they are in a way hinted at, but the merits of the scheme are now proclaimed in relation to a possible use of the liquid-air process as a means for the production of cheap oxygen gas.

Air, as of course all know, is substantially a mixture of four parts of nitrogen with one part of oxygen, and when liquefied in the usual way this proportion is substantially maintained in the liquid. As the liquid evaporates, however, the nitrogen escapes

much faster than the oxygen, and thus after a time what is left will be mainly oxygen, the nitrogen having escaped or been driven off, or, as we might say, distilled. If then we pass a stream of liquid air continuously into a vessel in which evaporation is allowed to take place, we will in time find said vessel to be filled with substantially pure oxygen.

If this was all and no effort was made to utilize the cold of the evaporation of the liquid air such a process would be very costly. Mr. Tripler's figure for the cost of liquid air, it will be remembered, was 20 cents a gallon, and five gallons of "liquid air" would be needed to produce one gallon of oxygen, or in other words, each gallon of liquid oxygen would cost one dollar.

This gallon of liquid oxygen would, on expanding to atmospheric temperature and pressure, give about 104 cubic feet, or, roughly, each cubic foot of oxygen would cost one cent, or a thousand cubic feet would cost $10.

It is, of course, obvious that by employing "regenerative processes" by means of which the cold from the escaping nitrogen and also that from the vaporizing oxygen was transferred to the air which was being liquefied, the cost of manufacture could be greatly reduced.

What that reduction would be there are, as yet, I believe, no data for computation, but it would without doubt be considerable so that it would seem probable that, as compared with the chemical methods heretofore in use for the production of oxygen, this new one would be decidedly cheaper.

The question which next presents itself is What shall we do with the cheap oxygen when we get it? This question is more difficult to answer. The present uses for oxygen are very limited, being chiefly for lime lights used in theaters and for stereopticon exhibitions and in some refined processes in metallurgy, such as the working of platinum. The amounts used for these purposes would obviously be quite inadequate to support a large business.

In view of this "the promoters" have naturally turned to the suggestion that oxygen might be used as a substitute for air in ordinary processes of combustion, as under steam boilers, in iron furnaces and the like. In this case the quantities required

would be amply great, but here another difficulty presents itself. The cheap oxygen will here encounter the yet cheaper "free" atmosphere as a competitor, and in that connection the cost of storage and transportation would be prohibitory.

For example, it has been suggested that if coal burning under the boiler of a steamship was supplied with oxygen in place of air there would be a saving due to the fact that the heat carried up the funnel by the nitrogen of the air would be retained if oxygen only was fed to the fuel. This saving has been, we think, incorrectly estimated at so high a figure as 40 per cent.

The most conclusive way to test the value of such a suggestion is to reduce it to a concrete case with actually calculated proportions of parts, volumes and weights, as for example, in one of our Atlantic steamers.

To begin with, every ton of such coal as is used on our ocean steamers would require 2½ tons of oxygen for its combustion,* and if this was carried along like the coal supply, the cargo capacity would be fatally reduced, as regards weight, even neglecting the strong steel cylinders in which the gas would need to be stored, and which would weigh much more than the gas.

As regards bulk also, even if the oxygen gas was carried under a pressure of 2,000 pounds to the square inch, each ton of coal would require about 500 cubic feet of compressed gas, and as 2,000 tons is a fair allowance of coal to carry one of our large steamers across the Atlantic,† this would mean that about 1,000,-

*One pound of carbon requires 2.66 pounds oxygen for its combustion, and 1 pound of hydrogen requires 8 pounds of oxygen.

English steamer coal contains about 80 per cent. carbon and 5 per cent. hydrogen; American steamer coal, about 75 per cent. carbon and 5 per cent. hydrogen. Average for both coals, say, 77 per cent. carbon and 5 per cent. hydrogen.

Oxygen for pounds of coal—

For carbon	$.77 \times 2.66$	$= 2.05$
For hydrogen	$.05 \times 8$	$= .40$
		2.45

or, say, 2½ pounds oxygen for combustion of each pound of coal.

†Tons of coal required for steamer across the Atlantic. Table in London "Engineering," 1893, Vol. I, p. 469, gives for the *Umbria* 1,900 tons. The estimate of coal for the *Campania* from the same table is 2,900 tons. The *St. Louis* has bunker capacity for 2,500 tons.—"Engineering," 1895.

000 cubic feet of compressed gas in cylinders would need to be loaded into such a steamer. This volume of gas alone, not allowing for any lost space between the cylinders, would be more than 10 times that of the "bunker" capacity for the 2,000 tons of coal.*

If the gas was not compressed, then each ton of coal would require about 66,000 cubic feet of gas, or for as much coal as would carry a steamer across the Atlantic 132,000,000 cubic feet. Now as the capacity of the huge gas holder which is such a conspicuous object in New York city as seen from the Hudson River is 3,500,000 cubic feet, it follows that no less than 38 such gas holders, or if one prefers, balloons of equal capacity, would need to be towed along by a steamer to furnish it with a full supply of oxygen. This, it is true, does not allow for the assumed 40 per cent. saving of fuel, and with that included only 22 balloons would be required.

To meet the above, it might be suggested that the oxygen should be manufactured as needed on board of the vessel. This, however, would require machinery comparable in bulk and weight with the engines driving the vessel, and, therefore, this suggestion also fails to relieve the difficulty.

Another practical difficulty which at once suggests itself to any one familiar with the combustion of fuels of any sort in oxygen, is the enormous intensity of the temperature produced. A furnace fed with oxygen in place of air would melt down or burn up in short order, not because a pound of coal would yield a greater quantity of heat if burned with oxygen, but because it would develop this quantity with a more intense combustion and so produce a greater and practically destructive intensity of temperature.

The above simple calculations show how utterly impracticable, not to say absurd, is the suggestion that oxygen might be used with advantage in the furnace of a steamer.

We have selected the case of a steamer from those referred to

* Space required to store a ton of coal, "Kent's Mechanical Engineer's Pocket-Book," p. 170. To store ton of coal, 2,240 pounds. Bituminous coal, 41 to 45 cubic feet. Anthracite coal, 34 to 41 cubic feet.

by the "promoters" of the new oxygen process, as involving data easily secured; but it goes without saying that the conditions with a locomotive or any other traveling motor would be substantially the same, and that even with stationary engines the bulk and mass of the oxygen gas to be handled would render its cost prohibitory by reason of the first cost and operating expense of the machines required to produce and manipulate it.—Henry Morton, Ph. D., in the "Steven's Institute Indicator."

THE DIAMETER OF PROPELLER SHAFTS.

The Technical Committee of the British Corporation for the Survey and Registry of Shipping, of which Mr. Archibald Denny, the well-known ship builder, is chairman, has recently had under careful consideration the important subject of the diameter of propeller shafts, in view of the large number of fractures, with more or less serious consequences, which have lately taken place.

The original rules of the British Corporation for shafting, as published in 1893, followed the general practice of that day, and specified that the propeller shaft should be of the same size as the crankshaft, the line shafting being 0.95 of the crankshaft. When propeller shafts began to give so much trouble, the Committee of the Corporation, being very much pressed by the shipowning and underwriting members, made an increase of five per cent. in diameter for the propeller shaft, and as the breakages continued, another five per cent. was added later. As more breakages were reported, the shipowners pressed the technical members to make a still further increase. During these constantly recurring breakages, however, the Corporation had practically no complaints as to the shafting of vessels built to their rules, a fact which is shown by an analysis of the lists drawn up by Lloyd's at the time. The Corporation rules required a slightly larger propeller shaft than Lloyd's, which might or might not account for the absence of complaints; but the fact remains that their percentage of fractures was exceedingly small. The technical members of the British Corporation Committee

decided before taking action to have a thorough investigation of the matter, with results which are embodied in the following new formula and rule, just issued; and it will be seen that it increases the crank and line shafting by 10 per cent. in strength:

1. *Diameter of Shafting.*—The minimum diameters of crank, thrust, propeller and intermediate shafts may be found from the following formula, except where the ratio of length of stroke to distance between main bearings is unusual, when they will receive special consideration:

$$D = \sqrt[3]{\frac{P.L^2.S.}{C}} \times B.$$

Where D = diameter of shaft.

P = absolute pressure, *i. e.*, boiler pressure + 15 pounds.

S = stroke of engine, in inches.

L = diameter of low-pressure cylinder, in inches.

B = 1.0 for crank and thrust shafts.

B = .95 for intermediate shafts.

B = for propeller shafts, to be taken from the following Table:

Coefficient of displacement of vessel at ⅔ molded depth.	Ratio of diameter of propeller to diameter of crank shaft.					
	13	14	15	16	17	18
.6	1.0	1.01	1.02	1.03	1.04	1.05
.62	1.01	1.02	1.03	1.04	1.05	1.06
.64	1.02	1.03	1.04	1.05	1.06	1.07
.66	1.03	1.04	1.05	1.06	1.07	1.08
.68	1.04	1.05	1.06	1.07	1.08	1.09
.70	1.05	1.06	1.07	1.08	1.09	1.10
.72	1.06	1.07	1.08	1.09	1.10	1.11
.74	1.07	1.08	1.09	1.10	1.11	1.12
.76	1.08	1.09	1.10	1.11	1.12	1.13
.78	1.09	1.10	1.11	1.12	1.13	1.14
.80	1.10	1.11	1.12	1.13	1.14	1.15

The value of the divisor C in the formula, depends on the ratio $\frac{L^2}{H^2}$ where L = diameter of low-pressure cylinder, and H of high-pressure cylinder in inches:

$\frac{L^2}{H^2}$ ratio.	Two cranks at 90 degrees compound or quadruple, also three cranks at 120 degrees quadruple expansion.	Three cranks at 120 degrees triple expansion.	Four cranks at 90 degrees quadruple expansion.
3⅛	9,910	.....	.....
3¼	10,160	.....	.....
3⅜	10,410	.....	.....
3½	10,660	.....	.....
3⅝	10,910	.....	.....
3¾	11,160	.....	.....
3⅞	11,410	.....	.....
4	11,660	.....	.....
4⅛	11,910	.....	.....
4¼	12,160	.....	.....
4⅜	12,410	.....	.....
4½	12,660	.....	.....
4¾	12,910	13,650	.....
5	13,375	14,160	.....
5¼	13,840	14,670	.....
5½	14,305	15,180	.....
5¾	14,770	15,690	.....
6	15,235	16,200	.....
6½	15,700	16,710	.....
7	16,630	17,730	.....
7½	17,560	18,630	.....
8	18,410	19,530	.....
8½	19,260	20,430	22,660
9	20,110	21,330	23,660
9½	20,960	22,200	24,660
10	21,750	23,070	25,660
10½	22,540	23,940	26,580
11	23,330	24,810	27,500
11½	24,120	25,660	28,420
12	24,900	26,500	29,340

2. The webs of built crankshafts should be keyed as well as shrunk on to the shaft, and the diameter of the shaft should be increased in the way of the web to make up the loss of sectional area at the keyway.

3. Propeller shafts should be forged from ingot steel or from rolled iron bars of good quality and not from scrap. Liners should be fitted in one length, but if not so fitted the space between the liners must be protected by efficient lapping. Joints in brass liners should be fused. Care must be taken to avoid abrupt changes of strength at any part of the length of the shaft.

New Clause in Rules for Special Periodical Surveys.—When

the after bearing is worn down to an extent exceeding two per cent. of the diameter of the shaft, it must be rebushed.

In connection with the first of the Tables, it may be said that in view of the fact that when vessels were moderately fine there were relatively few cases of broken shafts, the committee for their experiments took a fine ship with a moderate-sized propeller as being correct under the old rule, or nearly so, and started with a propeller shaft, the size of which was the same as the crankshaft—there is, however, the addition of 10 per cent. on the strength—then they reasoned as follows:

"The size of ships has greatly increased, while the speed and hence the size of shafts has not increased proportionately; on the other hand, owing to the size of the vessels, the propellers have increased in diameter out of proportion to the shaft, hence vessels of whatever fullness when at sea, if they have large propellers in proportion to the diameter of the crankshafts, while pitching and racing at sea bring increased percussive strains upon the shafts; and the committee decided to increase the size of the propeller shaft as the ratio of the propeller to the crankshaft increases. But on the other hand, any vessel of whatever size and with whatever size of propeller, if of very full form, will, when trading in ballast, have more of her propeller out of the water, which will bring greater percussive strains upon the propeller shaft than in a fine vessel. The committee combine these two reasons into one table as shown." Of course, no one can say that the actual sizes specified are correct, they must be more or less empirical, based upon general experience as to what has failed; but the committee consider that the idea is correct, and further experience will show whether the amount is sufficient. They think also that they are correct in going back to the old rule for fine vessels with small propellers, and on insisting that propeller shafts shall either be built of hemogeneous steel or first-rate scrap iron.

The second table is the same as the committee drew up at the beginning of the operations of the corporation, but the constants are modified to give the 10 per cent. greater strength.—"Engineering."

HAND AND MACHINE LABOR.

The following selection is taken from the series being published under this head in London "Engineering." It graphically illustrates the great advance made in a familiar field since the advent of modern machinery appliances.

Iron Pipe (Wrought).

This industry has been entirely revolutionized since the hand-making period, and, indeed, it may be regarded as a new industry. Table LXXV gives some general data on the manufacture of 100-feet run of 4-inch lap-welded pipe, and as the processes are different, we have given them in detail in Tables LXXVI and LXXVII.

Table LXXV.—Production of One Hundred Feet of Four-Inch Lap-Welded Wrought-Iron Pipe.

Mode of Production.	Hand.	Machine.
Date	1835	1895
Number of different operations involved	5	12
Number of workmen employed	3	59
Number of hours worked	84 h. 22 m.	4 h. 47 m.
Cost of labor	12.656 dollars.	0.850 dollars.
Average rate of wages per hour	15 cents.	18 cents.

Table LXXVI.—Production by Hand of One Hundred Feet of Four-Inch Lap-Welded Wrought-Iron Pipe.

Operation.	Appliance.	Time worked.		Rate of pay, per day.	Labor cost.
	Tools.	*h.*	*m.*	*Dols.*	*Dols.*
1. Beveling edges of plates	Blacksmith's	5	0	2.00	1.00
Ditto	Ditto	10	0	1.25	1.25
2. Heating, shaping, lapping	Ditto	7	30	2.00	1.50
Ditto	Ditto	15	0	1.25	1.875
3. Heating and welding	Ditto	10	0	2.00	2.00
Ditto	Ditto	20	0	1.25	2.50
4. Straightening	Ditto	2	30	2.00	.50
Ditto	Ditto	5	0	1.25	.625
5. Threading ends	Stock and dies	3	7	2.00	.625
Ditto	Ditto	6	15	1.25	.781

TABLE LXXVII.—PRODUCTION BY MACHINE OF ONE HUNDRED FEET OF FOUR-INCH LAP-WELDED PIPE (1895).

Operation.	Appliance.	Workmen.	Time worked.	Rate of pay.	Labor cost.
			minutes.	*dollars.*	*dollars.*
1. Beveling edges	Scarfing machine	One scarfer	7.2	0.235 per 1,000 feet.	0.0235
Beveling edges	Scarfing machine	One scarfer's helper	7.2	0.156 per 1,000 feet.	0.0157
2. Heating, shaping and lapping.	Bending machine and furnace	One bender	10	0.355 per 1,000 feet.	0.0355
Heating, shaping and lapping.	Bending machine and furnace	One bender's helper	10	0.249 per 1,000 feet.	0.0249
Heating, shaping and lapping.	Bending machine and furnace	Three bender's helpers	30	0.213 per 1,000 feet.	0.0639
3. Heating plate	Furnace	One welder's fireman	11	10.500 per week	0.0324
4. Welding edges	Welding machine and furnace	One welder	11	1.44 per 1,000 feet.	0.1440
Welding edges	Welding machine and furnace	One welder's helper	11	8.00 per week	0.0247
Welding edges	Welding machine and furnace	Three welder's helpers	33	7.50 per week	0.0694
5. Smoothing and polishing	Finishing machine	One rollman	11	7.50 per week	0.0231
Smoothing and polishing	Finishing machine	One engineer	11	7.50 per week	0.0231
Smoothing and polishing	Finishing machine	One inspector	11	8.50 per week	0.0262
Smoothing and polishing	Finishing machine	One barman	11	8.50 per week	0.0262
Smoothing and polishing	Finishing machine	One barman's helper	11	7.00 per week	0.0216
6. Straightening	Straightening machine	One straightener	12	0.445 per 1,000 feet.	0.0445
Straightening	Straightening machine	One straightener's helper	12	0.333 per 1,000 feet.	0.0334
7. Evening ends and threading	Cutting and threading machine	Threader	50	1.54 per 1,000 feet.	0.1540
8. Testing	Testing machine	Two testers	3.9	7.50 per week	0.0081
9. Upkeep of tools, etc.	Smith's tools	Two smiths	2.2	2.25 per day	0.0083
10. General	Trucks, etc.	Thirty laborers	14	7.00 per week	0.0259
11. Furnishing power	Engine	Two engineers	3	15.00 per week	0.0138
12. Firing boiler		Two firemen	3	9.00 per week	0.0083

From these tables it will be inferred that the machine process of making welded pipes is not only much more rapid, but is also far superior to the old hand method. At the same time the labor cost has been reduced from 50s. per 100 feet to less than 3s. 6d., although the operations of finishing and testing are included, details which did not appear at all necessary in the old days. Makers of welded pipe in this country will doubtless read with interest the analysis of American practice as set forth in Table LXXVII.

700-FOOT FLOATING DOCK AT SOUTH BROOKLYN.

The port of New York has hitherto been severely handicapped by the lack of proper drydock facilities for vessels of the larger class. It is well known that the fleet of mail steamers which plies between the old and new world includes the very largest vessels afloat; and while they start and arrive at a great many different ports in the old world, practically the whole of the fleet makes New York its destination on this side of the Atlantic. Whenever disaster has overtaken one of these big ships on its westward passage it has been necessary for it to go to some other port than New York for drydock repairs. It is satisfactory to know, however, that all this is being changed by the construction of a large floating drydock which will be located conveniently to the new 40-foot channel which is being dredged along the Brooklyn foreshore. This dock, which is to have an extreme length of 700 feet, will be capable of accommodating the largest vessel afloat, and the Oceanic, which has an over-all length of 704 feet, can be comfortably accommodated in the big structure. The drydock is merely one feature in an extensive and throughly up-to-date ship repair yard which is now being laid out and constructed by the Morse Dry Dock Co., at South Brooklyn.

The property is situated between Fifty-fifth and Fifty-eighth streets, and back on First Avenue, along which it extends for a distance of 426 feet. In addition to the floating drydocks there will be a wetdock or basin, the inner end of which will be about 100 yards from First Avenue, while the two piers which inclose

it will extend out 1,400 feet to the edge of the new Government 40-foot channel. The first 700 feet of the basin will be about 37 feet wide with a depth of 25 feet, and here vessels with a moderate draught of water will be berthed in the proximity of the machine shop. The outer 700 feet of the channel will be 100 feet wide, and will have a clear depth of 35 feet, or sufficient to accommodate the largest ocean liners. The pier to the south will be 22 feet in width, and that to the north 30 feet in width. On the north side of the latter pier will be the drydock basin, which will have a clear width of 130 feet and a depth of water at the entrance of 35 feet, while the length of this basin will be just 1,000 feet. The pier on the north side of this basin will also be 30 feet in width. The frontage of all three piers will aggregate 4,000 feet, and each pier will be traversed by a railroad which will run through the repair and boiler shops, and also have connection with other points in the yard. At about the midlength of the southerly pier there will be a large coal pocket for the accommodation of the ships which visit the basin.

Immediately to the east of the drydock basin will be a boiler shop, and beyond this a large two-story machine and repair shop, the width of each of these buildings being 80 feet, and their combined length 600 feet. The object of most popular interest is naturally the large pontoon drydock. This huge structure is not built, as might be supposed, as one integral structure, continuous throughout, but is composed of seven separate sections, which are identical in every respect. Each section is built of an approximate ⋃-form. The body of the pontoon is 120 feet wide, with a maximum depth of 15½ feet at the center. On each side are two tall wings, each measuring 56 feet from the bottom of the pontoon to the top platform, on which are located the motors for operating the electric pumps and handwheels for opening and closing the inlet valves. Each section is 80 feet in length, and, as there is a 36-foot extension of the floor of the two end sections, and a space of about 4 feet is allowed between the adjoining section, it will be seen that the total over-all length of the dock is approximately 700 feet. The dock is built of Southern pine, and a very complete system of stiffening bulkheads is

used, there being seven running longitudinally, or fore-and-aft through each pontoon. Each section is provided with a number of inlet gates below the water line, and it carries sufficient ballast to insure its sinking when the gates are opened.

On the floor of each wing are located two centrifugal pumps, each pair driven by a 50-horsepower electrical motor, which is located on the working platform above. These pumps have a capacity of between 5,000 and 6,000 gallons of water per minute, and it is expected that they will be capable of lifting a ship in from 30 to 45 minutes. The lifting capacity of each pontoon is 2,500 tons; consequently the combined lifting capacity of the whole dock, when coupled up, will be 17,500 tons. The repair buildings will contain the blacksmith shop, boiler shop, machine shop, and various other departments required in a modern plant of this kind. The machinery will be in every way up to date. Electric power will be extensively employed, most of the machines being run by direct-connected motors, and extensive use will be made of compressed air, not only in the shop, but throughout the yard. For the latter service lines of compressed-air mains will be carried down the full length of each pier, with valves at every 75 feet, to which flexible hose can be attached for work upon the ships themselves. This compressed air will be used for caulking, riveting, chipping, drilling, and the other operations incidental to ship repairs. The company estimates that it will be able to accommodate about 25 ships, big and little, at one time in the drydock and the adjoining basins.—"Scientific American."

THE ELECTRIC PROCESS OF ANNEALING ARMOR PLATE.

In "Notes" of our last issue we published a brief of Mr. Charles J. Dougherty's paper on the above subject which he had read before the Engineer's Club of Philadelphia.

As the full report is now available, we reprint the principal portion of the paper, in view of its general interest and the completeness of the description of the process.

Mr. Dougherty says:

For those not conversant with the "Harvey process" I will briefly state that it is the introduction of carbon by cementation into the face of an ordinary low-carbon steel plate, and subsequently it is water-hardened similar to an ordinary tool. After this treatment it presents a hard-faced surface to the depth of about 1 inch, designed to stop and break up projectiles before serious penetration takes place. All armor plates must necessarily be secured to the framework of the vessel in order to hold them rigidly. The plates must be drilled and tapped for this purpose, and herein lies the trouble. The methods heretofore used to produce isolated spots in the plates were principally two: (1) to protect the surface of the plate in patches or strips to prevent carburation wherever holes were expected to be drilled; (2) to make accurate drawings and patterns of each plate beforehand, to which all holes are drilled before the plate is hardened.

This plan was practiced by the United States government for a time, but had to be abandoned because it was found that numerous alterations in construction, errors in the draughting-room or mills, made it necessary to pierce the plate where no provision for annealing was made. It was also found feasible to drill and tap holes in the face of the armor plate at any stage of the process prior to hardening, and without detraction from the plate's resistance; but it is not always possible to locate these holes with precision without first fitting the plate in place on the ship; this was a very expensive method, and was abandoned in favor of one by which the carbon was prevented from penetrating over certain areas in the wake of the fastenings. This method also had its disadvantages, in that the carbon gases frequently seeped through the protection mentioned, and carbonized the surface beneath. The expedients adopted for preventing the carburation could not always be relied upon, and the great difficulty encountered was to pierce the surface, drill and tap such a face-hardened plate.

The first experiments which were made to anneal these face-hardened plates by means of the oxyhydrogen flame and the electric arc were not successful, and the plates resisted most effectually all attempts to anneal the spots which were required

to be drilled. Drills of every design and method of tempering were tried, but with no success; the plates could not be drilled by any of the above means. If the plates could not be held in position on the vessel, this meant serious delay in construction. The question of being able to anneal the spots required was becoming a serious matter; but as many times before in other work of difficult character, the subtile fluid was the only agent that solved the problem, so also in the case of taking the temper out of certain spots to permit the drilling of holes in the Harveyized plates, electricity came to our aid on the eve of despair and failure.

The Thomson Electric Welding Company, of Lynn, Mass., made some experiments on Harveyized plates, and very soon demonstrated their ability to anneal any surface, however hardened, by sending a current of large volume through any spot to be annealed, and by this means raising the temperature of the spot to about 1,000 degrees Fahrenheit; and at that temperature there can be little doubt that the temper has been withdrawn. This is the process that I wish to treat in this paper.

The apparatus which is necessary to perform the work of annealing these Harveyized plates consists of a separately excited alternating-current generator of variable potential, giving a maximum of 300 volts and a current of 100 ampères. The frequency of the generator is 50 cycles per second; the fields are separately excited by a small direct-current exciter of 110 volts and 20 ampères, made by the Thomson Electric Welding Company, and it is belted to the generator from a pulley on the commutator end of the shaft, and delivers to the field of the generator a maximum current of 15 ampères at full load. A German silver resistance box, placed close at hand under the control of the operator, varies the current in the generator fields, and thus increases or decreases the alternating electro-motive force at the brushes of the generator.

The annealing machine proper is an alternating-current transformer similar to the well-known welding transformers used in welding steel rails. For those of you who are not conversant with the meaning of the word "transformer" in an electric sense, it will suffice to say that an alternating transformer may be re-

garded as a species of dynamo, in which neither armature nor field magnets revolve as in an ordinary dynamo, but in which the magnetism of the iron circuit is made to vary through rapidly-repeated cycles of alternations, by separately exciting it with an alternating current. The primary coil of the transformer corresponds to the field-magnet coil of the dynamo, while the secondary coil of the transformer may be called the armature coil of the dynamo. In the alternating-current transformer, by whatever name called, the function of the iron core is to carry the magnetic lines of force (that are created by the current in the primary coil) through the convolutions of the secondary coil. The rate at which the magnetic lines due to the primary current are cut by the secondary circuit is the measure of the electro-motive force given to the secondary circuit.

With this explanation of a transformer we can understand better the workings of this annealing machine. The transformer is of the copper-clad type, that is, one in which the secondary is composed of two copper castings, each having a rectangular groove; when bolted together, these form a closed rectangular frame in which the primary coil is held. The hollow space intervening between the primary and secondary is filled with a heavy-body oil called "transil oil." This oil acts both as an insulator and a conductor of heat from primary to secondary. The secondary, by completely surrounding the primary, affords an excellent mechanical protection, and prevents electric as well as magnetic leakage. The latter features are quite important, since it is necessary to operate the annealer on board vessels in exposed positions during construction; yet it may be handled with immunity from electric shocks, even when operated in rain or in heavy snow.

The primary coil of this transformer consists of a copper ribbon, each turn of the coil being insulated from the next by thin asbestos paper. The ratio of conversion is 100 to 1, and when the maximum voltage of 300 is generated at the brushes of the generator, we have about 2.5 to 2.8 volts at the terminals of the secondary of the transformer. The transformer has two trunnions fastened to its sides in a line a little above the center of gravity.

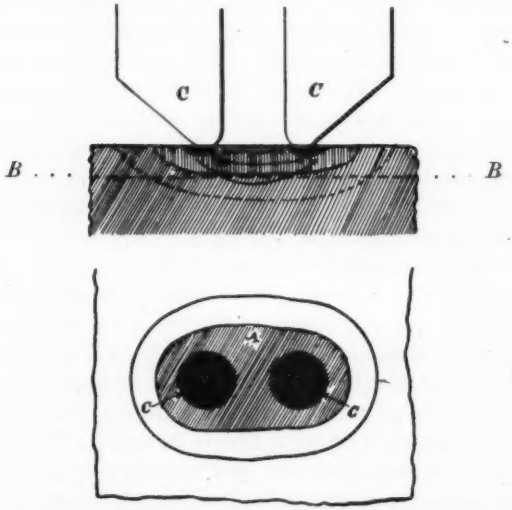
These trunnions swing in bearings—part of a yoke which straddles the whole. The yoke in its turn has a hook which may be secured at any place of the arch, thus allowing the transformer to be suspended, like a compass in gimbals, in any desired position. The copper castings which compose the secondary circuit are cut through at one place. On either side of the cut two short platforms form the base for a saddle which holds the copper contacts of various shapes and sizes, by means of which the current is made to enter and leave the plate to be annealed. These contacts are made of forged copper of high conductivity, and they are hollowed out to receive a water circulation to keep them cool, and they terminate in a narrow tip which is rounded at the end. The total weight of the annealing transformer is about 1,000 pounds, and this is sufficient to insure proper contact pressure for all work on horizontal plates. When inclined surfaces are to be worked upon, the machine is suspended so that its weight shall not interfere with the contact pressure, which is then obtained by bracing the contacts directly with wooden wedges against any object near by. Perhaps the most remarkable thing in the operation of annealing a spot is the great amount of current that is carried by the copper contacts into the plate. The contact surface on the plate is seldom more than about half an inch square, yet 10,000 ampères are made to flow through it during the process of annealing. This current density per contact area is equivalent to 40,000 ampères per square inch, a density which is only rendered possible through the thorough cooling of the copper contacts by a continuous water circulation through them, as I have before mentioned.

In the original experiments made by the Thomson Company in annealing plates, one feature presented itself in the fact that taking off the heating current when the spot was being annealed caused the heat to be so rapidly conducted away by the surrounding mass of metal that the heated spot became chilled, just as if it had been plunged into cold water. No manner of outside protection of the heated spot would prevent this chilling effect, and it was found that only a gradual and slow withdrawal of the current would produce the complete effect of annealing.

The current must be brought up to the maximum value and then gradually cut down.

The interesting operation of annealing a plate is performed in the following manner: The transformer is placed in position, the contacts touching the plate on either side of the spot marked for annealing. Then the primary current is brought up by means of the rheostat, near at hand, to from eighty-five to ninety-five ampères for a period of from four to five minutes. The metal between the two contact places soon attains a dull red heat, and this temperature is experimentally found by holding a small pine stick in contact with the spot until it takes fire. This is the maximum temperature desired to anneal the spot properly. The current is now gradually diminished by turning the rheostat handle one point each minute until all the resistance is placed in circuit, and by this method the spot is gradually cooled and the chilling of the plate prevented.

To illustrate the method of introducing the annealing current into the plate, I shall refer to the figure. *CC* are the two copper



METHOD OF INTRODUCING THE CURRENT INTO THE PLATE.

electrodes, the current enters the plate by one end and leaves by the other, as shown by the arrows. Right under the contacts the metal comes to a bright cherry heat, shown in heavy black in the figure, while the portion intervening and partly surrounding the contacts acquires a temperature of just visible red. Line *BB* indicates where the influence of the Harvey process stops. The shaded portion in the figure shows the zone softened and ready to be drilled, while the dotted line shows how far the heat radiation would cause the metal to turn blue.

The operation of running a heat after the machine is set up takes in the neighborhood of from fifteen to twenty minutes, all depending on the size of the spot to be annealed. On examining the spot after the annealing process is finished, it is found to be a dull chocolate color, the place where the contacts have been resting is scaled and hard, and cannot be touched by a tool to the depth of a quarter of an inch.

In the construction of a modern man-of-war there are many armor plates which act as shields to the guns. Some of them are circular, others oval. The only method possible to perforate these shields was after carburation and before being water hardened, up to the discovery of the electric annealing process above described; but to-day, what a change! These plates are cut in various shapes to suit the work by simply annealing a number of spots forming the shape of the hole to be cut, and finally running a cutting tool over the surface thus annealed.

In cases when the transformer is moved relatively to the plate to be treated, it is a rather startling thing to find a number of steel chips lying around, which have been cut from the plate by the copper contacts; thus developing the peculiar phenomenon of a hard steel chip cut by a soft copper tool.

Besides annealing holes in armor plate, the transformer may be used for the reversal of the annealing process, namely, for creating isolated hard spots in soft tool steel by sending a current through the spot to be hardened until it reaches a bright cherry heat, and then suddenly removing the current from the machine.

For annealing plates for the U. S. S. *Alabama*, two complete

annealing plants have been at work continuously night and day, Sundays included, for a period of over six months, and upward of about 3,000 holes have been treated.

We have had no difficulty in annealing the Harveyized plates of the *Alabama*, but in this age of advancement something new is constantly being invented; yesterday it was Harveyized nickel-steel, and to-day Krupp's armor plates are considered superior to the Harvey or Corey plates. Krupp armor is now being placed upon all the newer vessels of our own and foreign navies.

The Krupp gas process, as it is called, is a secret, and the rights of manufacture for this country have been purchased by the Carnegie Company, of Pittsburg.

The supreme value of the Krupp armor consists in the fact that these plates enable the total weight of armor with which a ship is clothed to be greatly reduced, and the weight so saved may be appropriated to a more powerful armament, or engines and boilers of greater horsepower. Krupp armor shows remarkable toughness combined with all the hardness of the best face-hardened armor; and, unlike armor manufactured by other well-known processes, the Krupp product maintains these qualities in the very thickest armor. The thickness of the hardness of Krupp's process is about 1.7 inches to 2 inches, while the Harveyized process goes in about 1 inch. The new battleship *Maine* will have Krupp armor; the Russian battleship and cruiser now building at our shipyard will also be protected with it.

Since the Krupp process penetrates into the surface of the plate about 2 inches, we naturally imagined that it would require a greater length of time to anneal the spots, and after a little experimenting on a sample of this armor we were delightfully surprised to find that the time required to anneal a hole thoroughly was only a trifle longer than that for the Harveyized steel plate.

MECHANICAL STOKERS.

The desirability of substituting mechanical stoking for hand labor is a matter about which there is little difference of opinion, the principal questions arising being those of cost, expediency

and adaptability. The subject is so often complicated with matters not directly belonging to it that the paper of Mr. W. E. Snyder, recently read before the Engineers' Society of Western Pennsylvania, and published in the *Proceedings* of the Society, is altogether welcome, treating as it does of the furnace considered apart from the boiler, as a device for the efficient combustion of fuel.

Ordinarily the performance of a furnace is complicated with that of the boiler with which it is operated, the efficiency being considered as measured by the evaporation, regardless of the fact that the evaporation is largely affected by the character of the boiler and of its setting. Mr. Snyder, on the contrary, seeks to find an expression for the efficiency of the furnace itself as a means of comparing various methods of conducting combustion, realizing that it is absurd to measure the efficiency of a device for generating heat by the ability of an entirely independent apparatus to absorb that heat.

In endeavoring to establish an independent standard of furnace-efficiency Mr. Snyder assumes that the perfect furnace delivers only gaseous products of complete combustion to the chimney, and only incombustible earthy matter to the ash pit, and uses no more air than is sufficient to supply the oxygen to unite with the oxydizable components of the fuel. By measuring the losses which occur by passage of unburned fuel to the ash pit, unburned gases to the chimney, and excess air admitted and carrying away heat, we are able to determine the extent to which a furnace departs from this perfect performance, and so obtain an efficiency for it entirely apart from that of the boiler to which it may be attached. Mr. Snyder gives several examples of this method of determining the efficiency of furnaces and shows how it enables the engineer to study the real value of a given furnace. If the furnace loss is low, indicating good combustion, but a low evaporation is shown by good coal, no blame should be placed on the furnace. The heat generation has been proven efficient; the responsibility for its efficient absorption lies with the boiler.

Comparing different methods of firing, according to the above idea of furnace efficiency, the defects of the common grate are

very apparent. It permits some coal to fall through into the ash pit, and it requires a large excess of air to enter when firing, while at the same time the chilling of the furnace prevents complete combustion, so that the losses at all the points are necessarily large. Instead of providing the essentials to good furnace economy, *i. e.*, high furnace temperature and proper air supply, there is a fluctuating furnace temperature and large volume of excess of air at times. The defects of the ordinary grate are not diminished by the use of the various shaking or other grates of special design, these only serving to assist the fireman in the work of cleaning, and in no way removing the great defect of intermittent firing through open doors.

Various principles of design have been adopted in the attempts to make a mechanical stoker which shall obviate the defects of hand firing, and apart from the saving of manual labor the object which they all have in view is the uniform and continuous supply of the fuel, together with a properly regulated supply of air, so that as perfect a combustion as possible may be secured.

The mechanical devices used in stokers may be of three kinds: One acts to spread the fuel upon the fire, another feeds it at one end, and the third feeds it up from beneath. In each of these the gaseous products are supposed to be fully consumed, either by passing over the incandescent fuel, or by passing through the bed of burning coal.

The first class is represented by several makes used in England and on the Continent, but is little used in America, where the second type, with inclined grates, is more common. In both of these good combustion can be secured, and a number of successful installations have been made, but there are certain defects which act to prevent the highest attainment of efficiency. Although the combustible gases produced by the coking of the coal are passed over the incandescent bed, which is at a temperature probably as high as 2,500 degrees Fahrenheit, yet they are also obliged to pass under and near the surface of the boiler, which is not hotter than 400 degrees Fahrenheit. The result is that only a partial combustion is attained, and some unburned gases go out at the chimney. The action of the open and moving grate bars

of such stokers is also defective in that much fuel falls through unburned into the ash pit. Nevertheless such stokers are doing good work in many places, and when their efficiency is considered apart from that of the boilers under which they are placed their superiority to the ordinary grate is evident.

When the fire is fed from beneath, as in the third class above mentioned, these difficulties are modified. The combustible gases arising from the distilled coal have to pass through the fuel bed; and, if the proper quantity of air is present, they will be fairly well burned, although there must doubtless be a portion of the chilling effect of the boiler surface yet present. The solid cast floor prevents the loss of fuel, and this feature renders this type well adapted for burning inferior and finely divided fuels, so that the choice of a stoker should in a measure be governed by the character of the fuel with which it is to be used. The nature of this type of stoker renders forced draft necessary, and this feature enables the rate of combustion to be controlled by regulating the draft, while the high rate of combustion thus attained gives also a high temperature and permits a decrease in the amount of air per pound of coal.

It is evident that the laws of combustion must be observed in the design of a stoker if good results are to be obtained. The extent to which these laws have been observed can be determined by the character of the solid and gaseous products of the furnace, and when the efficiency of the furnace is determined from these data results are obtained which are comparable with those obtained in a similar manner from other furnaces. There is no doubt that in all cases mechanical stoking, properly conducted, is better than hand firing, but it must not be expected that careless usage and neglect will conduce to good results. A higher grade of attendance is given to a first-class Corliss engine than to a common, plain slide-valve engine, and what is good sense in the engine room is also good sense in the boiler room.—"Engineering Magazine."

PROGRESS OF THE METRIC SYSTEM IN THE UNITED STATES.

In printing an abstract of the report of the Committee on the Metric System, presented at the Chicago meeting of the Ameri-

can Railway Association, April 25, 1900, attention is called to an article upon the Metric System contributed by Dr. Coleman Sellers to "Cassier's Magazine" for March, 1900.

In this article Dr. Sellers sets forth the anti-metric arguments very lucidly. To the Naval Engineer, however, these may not appeal with the force they do to the manufacturer, as in a profession which affords so many opportunities for consultation with, and criticism of the work of, foreign engineers, the adoption of an international metric standard has many features of advantage in the line of progress.

While not, therefore, losing sight of this, we have appended to the below-given report some interesting extracts from Dr. Sellers' article as being pertinent to the findings of the committee as well as explanatory of the lack of enthusiasm for the change existing among American Mechanical Engineers.

Report of Committee.—In conducting the investigation necessary for this report, the fact was early made apparent that the use of the metric system in commercial lines in the United States seemed to be confined to comparatively few and isolated cases of manufacturing companies, engaged in the export of their products to foreign countries wherein the metric system prevails; such being the case, your committee has not attempted to locate and report upon all of such cases, nor to specify them particularly. This fact has made it necessary to make this report much more general than might have been desired or expected. The use of the metric system in the United States was legalized by Congress in 1866. It was provided thereby that the use of the metric system in all contracts, etc., did not render such invalid, and thereby made the use of it optional and legal, but not compulsory. Various legislative and committee work has since been done in response to memorials presented to Congress by different associations and scientific societies throughout the United States.

In December, 1895, a bill was introduced into the House of Representatives by Mr. Hurley, which was referred to the committee. At the first session of Congress in 1898 Mr. Hurley again introduced a bill to fix the standard of weights and measures by the adoption of the metric system. This was

referred to the Committee on Coinage, Weights and Measures, since which time nothing definite has been done with it. The bill previously introduced by Mr. Hurley did not make the use of the metric system compulsory upon the people, but simply established it as a legal standard to which reference could be made in cases of dispute, or for any other proper purposes. On March 16, 1896, a bill was introduced into Congress, providing as follows:

(1) From and after July 1, 1898, all departments of the United States Government, in the transaction of all business requiring the use of weights and measurements, except in completing surveys of public lands, shall employ and use only the weights and measures of the metric system.

(2) From and after January 1, 1901, the metric system of weights and measures shall be the only legal system of weights and measures in the United States.

The bill was debated April 7, 1896, recommended April 8 to the same committee, to be brought forward in the 1897 session. Nothing further has thus far resulted from this move.

Your committee finds that there has been no legislation in any of the States regarding the use of the metric system, except in a few cases which provide that the metric system may be employed, and that no contract shall be invalid because of its use therein, covering practically the same ground covered by the United States Congress when legalizing the system in 1866. In the work of the Smithsonian Institution the metric system is used almost exclusively.

In the different departments of the United States Government at Washington the metric system is used as follows: In the Department of the Coast and Geodetic Survey the metric system is used exclusively in their computations. Their charts, while they show soundings in feet, show also a metric scale, by the use of which the different dimensions shown in feet may be readily transformed into metric measurements. In the Agricultural Department all scientific work in chemistry, etc., is done by the use of the metric system. The extensive natural history work in this department is also carried on with the use of the metric

system exclusively for its calculations. The Post Office Department uses it in connection with foreign metric countries, but not in any work applying to our own country exclusively. In the Department of the Surgeon General of the Army their contracts for medical supplies all embody the metric system, and their supply table (page 82, Manual of 1899,) shows all supplies as being packed in metric packages. All of the Army post physicians and apothecaries are required to use the metric system in making and filling prescriptions. The Department of the Surgeon General of the Navy also uses the metric system in much the same way as that Department of the Army, and manufactures its own bottles and packages in accordance with metric measures. When supplies are purchased for the Navy Department in the original packages, these supplies are repacked in the metric packages furnished by the Navy Department. Article 670, Regulations for the United States Marine Hospital Service, 1897, reads as follows: "Officers shall, for all official, medical and pharmaceutical purposes, make use of the metric system of weights and measures. In expressing quantities by weight, the term 'gram and certigram,' and in expressing quantities by measure, the term 'cubic centimeter' only shall be employed." In Porto Rico and Cuba the metric system has been and is exclusively in use in all official and domestic work, the natives and residents of those countries being familiar with no other system.

Manufacturers of measuring implements, gages, instruments of precision and draftmen's supplies make metric instruments for such uses. A few manufacturers of firearms, screw threads, screw-cutting lathes, manufacture machinery in metric dimensions for export to foreign countries, but not for use in the United States. A Philadelphia firm and a New York firm make injectors to the metric scale; a Massachusetts and Rhode Island firm make drills and gages; one of the largest scale manufacturing companies manufactures metric scales, and one of the largest watch manufacturing concerns in our country has used metric measures since 1869. Of the large steel companies, none use the metric system, except when called for by foreign export trade. The system is used almost entirely in the manufacturing of electrical

goods, and one refrigerator manufacturing company uses the metric system in everything that it manufactures, except fittings.

The Solvay Process Co., of Syracuse, N. Y., manufacturing soda ash and bi-carbonate of soda, uses the metric system in every possible way in its extensive works. It is not used for lineal measurements. All quantities or sub-division of coal or other supplies; the measurements of liquors or water; the calculation of heat units, and the measures of temperatures are all specified under the metric system. The drawings to go outside of their works for construction, etc., are not made in the metric system. The metric system is exclusively used in the laboratory and in all chemical work. A number of other isolated cases have been brought to the attention of your committee wherein the metric system is in use by individual manufacturing concerns, but it is not thought necessary to detail them, as they are comparatively few. Prominent manufacturers of astronomical and physical instruments use the metric system in their measurements of parts to replace other parts in instruments of precision, where such instruments of precision have been manufactured in metric countries. In electrical engineering and experiments the metric system is almost exclusively used.

As the result of the investigations that your committee has been able to make, its conclusion is, that thus far the introduction of the metric system in ordinary commercial lines of the United States has been very gradual, and generally confined to cases in which manufacturing concerns export to metric countries machinery made by metric scales for use in such countries. Except as herein noted, your committee is unable to find that the system is being introduced into this country to any considerable extent, in the way of supplanting our present system of weights and measures, for our own use. The theoretical study of the metric system is made a part of the course of instruction given in our schools, colleges and universities.—"Engineering News."

EXTRACT FROM DR. SELLERS' ARTICLE ON THE METRIC SYSTEM.— * * * To the great bulk of mankind engaged in trade, in buying and selling, in bartering and exchanging, it

matters little what system of weights and measures they adopt; it matters little whether they are obliged to use a yard stick or a meter rod, pounds or kilogrammes, quarts or liters. The cost to them of a change from one to the other is the cost of the few devices needed in weighing and measuring; the *rationale* of the system may never enter their thoughts.

With the machinist the case is different. He must not only possess costly means of measuring and weighing, with a degree of exactness unknown to others, but the results of these weights and measurements are fixed and unalterable. Enormous expenditures on tools, on drawings, on patterns, on everything he uses in making or building his machines are on what is involved in the primary system used in determining weight and size. The product of this expenditure means everything that makes modern civilization possible. I propose, in this paper, to consider the subject only as it relates to the engineering profession; not in regard to its effect on the grocer, the dry-goods man, or on the druggist. I propose to show why, afters many years' constant use of the metric system of measurement, I record my opposition to any enforcing legislation in this direction, because the metric system is not well adapted to the practice of the machine shop.

* * * * *

To show the measure of the misapprehensions as regards its effect on the engineering profession, on the part of the enthusiastic advocates of the exclusive metric system, I will pass by some lesser lights and seek illumination from the central luminary.

Professor Frederick A. P. Barnard, S.T.D., LL.D., who was President of the American Metric Bureau, and of the American Metrological Society, said, in December, 1877: "It is now little more than a dozen years since the movement in favor of the reform of the confused metrological system of the United States was set on foot. Originating with a few '*advanced thinkers*' (the italics are mine) and regarded with indifference by the multitude, it encountered, as it is the fate of all efforts to emancipate mankind from the burden of traditional evils to encounter, a much

larger degree of opposition than of encouragement." Then, after denouncing many of those who oppose the forced introduction of the system, he says:

"Among the arguments urged by those who maintain the impossibility of change only one appears to have much force, and that is the argument drawn from the dependency of machinery upon minute exactness in the measurements of parts, and from the great expense which would attend the adaptation of machine shops and machines to a new system." Quoting the majority report of the Franklin Institute as to probable cost, and referring to the second report of the committee of the New York University Convocation, in which report the number of dimensions requiring separate indications on the drawings of a 25-horse-power steam engine are given, he continues: "Singularly enough these statements, and all the rest of the same class in both the reports referred to, instead of being arguments against the abolition of the present metrological system and the substitution of the metric for it, afford the strongest reason for believing that that is precisely the thing which ought to be done. We desire, I suppose, to create a demand for our steam engines and our manufacturing machinery on the continent of Europe. * * * But a steam engine or a machine, all of whose parts are measured in English linear measures, if transferred to a metric country and there, by accident, disabled, becomes nearly useless, *since the shops of such a country afford no facilities for repairing it*," etc. (the italiacs again are my own).

It seems needless to tell engineers that these statements show so entire an ignorance on the part of Professor Barnard of the merits of the case as can scarce be credited from such a source. I feel ashamed to tender to him an explanation of what is involved.

The unit of measurement used in making a machine does not in any way complicate the repairs of that machine. Machines built in Great Britain do not always agree with any of the even sizes in the United States; yet this discrepany is a matter of no moment in the repairs of any of them. If we fail to find sizes corresponding to American sizes in British machines, presumably built on the same scale of linear measurement as the American

ones, neither do we find even millimeter sizes always in machines from France or Germany.

* * * * *

The metrology of the American shops is based on the inch, and on it only. This dimension is cut up into minor parts by halving to any degree of subdivision, practically in shop sizes to $\frac{1}{16}$; it is also divided into 10 parts and into 12 parts when such divisions serve any good end. All such divisions have their uses and lead to no confusion. The inch squared is the base of our strains and pressures. The inch cubed gives us capacities. Later, I will speak of 12 and 36 inches squared and cubed. This one unit, the inch and pound weight of 7,000 grains Troy, is all that a machinist needs to carry on his business. His inch is the same inch as is used in Great Britain and in the Russian machine shops. His pound is the pound in common use in Great Britian.

In America we have dropped some needless weights and measures. We do not use in the machine shop the ton of 2,240 pounds nor its quarter or its hundred weight; we do use a weight called "ton of 2,000 pounds." This is the factor weight in strains, and by it we sell machinery. Other trades may retain some of these useless things; I am speaking only of machine-shop practice.

The unit of measurement in France and in Germany is the millimeter. It is not and cannot be the meter for the following reason:—The great majority of all sizes used in the construction of any machine, whether it be big or little, are less than one meter. By the use of the millimeter only, decimals are avoided. Eight millimeters must be written 8 in the millimeter scale, while it must be written 0.008 in the metric scale. This is reason enough; for by the use of millimeters only, confusion of signs is avoided, and the danger incident to decimals is avoided; hence all drawings are figured in millimeters only, up to dimensions measuring many thousand millimeters. This little dimension is then squared and cubed, or ten or one hundred millimeters are squared and cubed, for the uses corresponding to the squared and cubed inch and foot.

As may be expected, happy coincidences of conveniences are

found in either system. Thus, an ardent metric advocate instances that 1 kilo to the square centimeter is just one atmosphere. We say 15 pounds to the inch is an atmosphere. Neither one is right, but the 15 pounds to the inch is 1 per cent. nearer right than the other is. For a machinist who seldom uses atmospheres a happy coincidence on the other side will be of more service. It so happens that wrought-iron bars with parallel sides measure in square inches of this section just $\frac{1}{10}$ of their weight in pounds per yard. Now, inasmuch as "shapes" in iron are rated by the pounds per yard, for convenience in large structures, so it comes to pass that when we know the weight per yard of any wrought-iron "shape" we know at once its sectional area. Inasmuch as compression and extension and factors of safety are involved in a knowledge of cross-section, it is handy to be able to find it so readily, is it not? A shape iron, 80 pounds to the yard, has 8 square inches area of section. If it is good for 10,000 per square inch in extension we may load it with 80,000 pounds.

I have set out to compare the two scales—to compare the two units, rather—after an experience of many years with both. The inch is 25.4 times larger than the millimeter. These are the two dimensions we are to compare.

We will begin in the drawing room. Here "the inflexible laws of mechanics" find their first expression in form on paper. Few machines, or even parts of machines, can be drawn full size. Hence comes the need of "scales." There is reason in all things, even in scales. The unwritten law of most machine shops is to make every drawing as large as possible, as near full size as the nature of the subject and the dimensions of the paper used will permit. We have in our drawing room about 125 drawers, each of which will take in, without folding, drawings 52 inches (1,320 millimeters) long by 33 inches (840 millimeters) wide. This is about as large a sheet as we can use to advantage, and tracings from these are not unmanageable in the workshop.

For metrical drawings we can use the following scales only:

Full size in which 1 mm. = 1 mm.
One-half size in which $\frac{1}{2}$ mm. = 1 mm.
One-fifth size in which 2 mm. = 1 cm. or 2 cm. = 1 dm.

One-tenth size in which 1 mm. = 1 cm. or 1 cm. = 1 dm.
One-twentieth size in which 5 mm. = 1 dm.
One twenty-fifth size in which 4 mm. = 1 dm.
One-fiftieth size in which 2 mm. = 1 dm.

Down to the one-tenth scale the dimensions can be read from a good millimeter rule; for the one-twentieth, the one-twenty-fifth and the one-fiftieth scales must be constructed. The jump from one-half to one-fifth size is unfortunate. Could we conveniently quarter the whole size we would have an increased area section, a matter of much moment. One-fifth of 10 inches is 2 inches, and the square of 2 is four. One-fourth of 10 inches is $2\frac{1}{2}$ and its square is $6\frac{1}{4}$, a gain in size of over 50 per cent.; a gain in comfort, in convenience and in eye-sight. Here we catch the first glimpse of the advantage of our own system; for with it a draughtsman can, from an ordinary well-divided inch rule, obtain the following scales: Full, $\frac{1}{2}$, $\frac{1}{3}$, $\frac{1}{4}$, $\frac{1}{6}$, $\frac{1}{8}$, $\frac{1}{12}$, $\frac{1}{16}$, $\frac{1}{24}$, $\frac{1}{32}$, $\frac{1}{48}$, $\frac{1}{64}$,—12 gradations, as compared to 7, and to these 12 can be added with perfect ease 5 of the others, making 17 in all, if the preference be for the decimally-divided inch, a scale carried in the tool box of every machinist, and obtainable from the two-foot rule in so common use.

The scale series in most common use is that of $\frac{1}{2}$, $\frac{1}{4}$ and $\frac{1}{8}$; this halves down from whole size and can be raised, in rapid drawing, by taking off diameter sizes from one drawing and using them as radius dimensions in the other, a process impossible between $\frac{1}{2}$ and $\frac{1}{5}$ sizes.

The true value of this extended series of scales, with its peculiar advantages, is manifest to any one familiar with both, and admits of no dispute. Is it a wonder that draftsmen brought up under a metric rule take so kindly, as they do, to our unphilosophical system?

Drawing is but a small part of the engineer's work. More or less calculating has to be done; many hours must be spent in figuring strains, estimating weights, determining speeds and what not. This brings us to the test of convenience in calculation, to the stronghold of the metric advocates. It is just here that Dr. Edward Wigglesworth, in his metric tracts, comes out

the strongest in his peculiar style. He says that Americans, self-ruling, are really too lazy, while merely claiming to be too stupid, to use the system. He says, "Shame on a country which to party gives up what was meant for mankind." It is claimed that the decimal notation of the metric rule gives greater facility in calculating, but that this is not its sole advantage.

Mr. Frazer says:—"Let the carpenter or mason be asked how many tons of water a structure, whose external (he probably means internal) dimensions are given, by his rule, will contain, and they will acknowledge that the decimal division is not the only advantage of the metric system, but that another is the perfect relation of extension, capacity and weight." For a structure 5 feet square and 10 feet deep the carpenter would divide, in his head, 250 by 32 and say 8 tons nearly, or 7.8125 tons exact, of the tons of 2,000 pounds in use, in all such measurements. The metric system would give for a nearly similar structure, say $1.5 \times 1.5 \times 3$ meters, a result obtained by tedious multiplying only; but what then? This problem applies to water only. If these spaces were to be loaded with bricks the metric multiplication must be still further multiplied by the specific gravity of bricks, thus:—$1.5 \times 1.5 \times 3 \times 1870 = 12{,}662.5$ kilos, while our mason's sum would read $5 \times 5 \times 10 \times 125 = 31{,}250$ pounds of common hard bricks, with an ease of calculation rather in favor of the two-foot rule.

I have mentioned the innumerable books which have been prepared, simplifying processes of calculations by tabulating the results of experiments on the basis of the inch unit. Of these books the British experiments form a large bulk of the valuable engineering knowledge of the world. Hodgkinson, for example, experimented with the crushing resistances of various substances, and the result of his experiments are in the possession of all engineers. He took samples in cylindrical form 1 inch diameter, 2 inches long each, for these experiments. Armengaud quotes these experiments and tabulates the results, saying they were obtained by Hodgkinson, *avec des cylindres de* $0^{m}0254$ *de d*^{tre} *sur* 0808 m. *de haut*$'' = 1'' \times 2''$, and from these he deduces, for example, that ash has a crushing resistance of from 610 to 653 kilos per square centimeter.

Mr. Trautwine, quoting Eaton Hodgkinson's experiments, also tells us that ash, weighing from 45 to 53 pounds per cubic foot, has a crushing value of 8,600 pounds per square inch. Now, 1 pound per square inch = .073077 kilo per square cm., or 1 kilo per square cm. = 14.2232 pounds per square inch, $\frac{8{,}600 \text{ pounds}}{14.22}$ = about 605 kilos to the square cm. Here, from Trautwine's deduction, the metric-using engineer will employ 605 as a factor where we use 8,600 in the same case. He, because his unit of measurement is less, or, rather, requires more figures to express it, multiplies these many figures by a lesser factor, while we, expressing our dimensions with lesser figures, use with these figures a larger factor. In other words, we can complete our calculation sooner because we are able to deal with the largest measures compatible with convenience. We can use the cubic inch, the cubic foot or the cubic yard, at our pleasure, just as the mechanic selects his tools in accordance with the extent of his work, and does not waste time driving at a railroad spike with a tack hammer.

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The value of the drawing-room system is tested or tried when the drawings reach the machine shop. It is there that errors are found out. An incorrectly-figured drawing costs nothing on account of the errors so long as that drawing rests quietly in its drawer; but it costs fearfully when the error is discovered in the partially finished machine. All engineers agree on one thing, viz: the fewest possible figures that can be used to express dimensions clearly, the easier it is to work to the drawing, and the less liability to make mistakes. Beautiful as is a decimal system in calculation, and we all use it, save in mental arithmetic, it has been found advisable to avoid the use of decimals as far as possible on the drawings used in workshops, even in metric-using countries. A misplaced point is an easy error to make, and may cause no end of trouble and expense.

* * * * *

The first item of manufactured matter entering the machine-shop door is bar iron. The merchant sizes of round, square, etc.,

WHITWORTH'S SCREWS.—IN USE IN THE UNITED KINGDOM AND EUROPE, ESPECIALLY IN GERMANY.

Diameter. English inch.	Diameter. Prussian inch.	Diameter. Millimeter.	Diameter at root of thread. English inch.	Number of threads per diameter.	Number of threads per inch. English.
$\frac{1}{4}$	0.243	6.4	0.186	5	20
$\frac{5}{16}$	0.303	7.9	0.241	$5\frac{1}{3}$	18
$\frac{3}{8}$	0.364	9.5	0.295	6	16
$\frac{7}{16}$	0.425	11.1	0.346	$6\frac{1}{3}$	14
$\frac{1}{2}$	0.486	12.7	0.393	6	12
$\frac{9}{16}$	0.607	15.9	0.509	$6\frac{1}{2}$	11
$\frac{5}{8}$	0.728	19.0	0.622	$7\frac{1}{2}$	10
$\frac{7}{8}$	0.850	22.2	0.733	$7\frac{1}{3}$	9
1	0.971	25.4	0.840	8	8
$1\frac{1}{8}$	1.092	28.6	0.942	$7\frac{7}{8}$	7
$1\frac{1}{4}$	1.214	31.7	1.067	$8\frac{1}{4}$	7
$1\frac{3}{8}$	1.335	34.9	1.162	$8\frac{1}{4}$	6
$1\frac{1}{2}$	1.457	38.1	1.287	9	6
$1\frac{5}{8}$	1.578	41.3	1.369	$8\frac{1}{2}$	5
$1\frac{3}{4}$	1.700	44.4	1.494	$8\frac{1}{2}$	5
$1\frac{7}{8}$	1.821	47.6	1.591	$8\frac{7}{16}$	$4\frac{1}{2}$
2	1.942	50.8	1.716	9	$4\frac{1}{2}$
$2\frac{1}{4}$	2.185	57.1	1.930	9	4
$2\frac{1}{2}$	2.428	63.5	2.180	10	4
$2\frac{3}{4}$	2.671	69.8	2.384	$9\frac{5}{8}$	$3\frac{1}{2}$
3	2.913	76.2	2.634	$10\frac{1}{2}$	$3\frac{1}{2}$
$3\frac{1}{4}$	3.156	82.5	2.857	$10\frac{9}{16}$	$3\frac{1}{4}$
$3\frac{1}{2}$	3.399	88.9	3.107	$11\frac{1}{2}$	$3\frac{1}{4}$
$3\frac{3}{4}$	3.642	95.2	3.323	$11\frac{1}{3}$	3
4	3.885	101.6	3.573	12	3
$4\frac{1}{4}$	4.127	107.9	3.805	$12\frac{7}{16}$	$2\frac{3}{4}$
$4\frac{1}{2}$	4.370	114.3	4.055	$12\frac{7}{8}$	$2\frac{3}{4}$
$4\frac{3}{4}$	4.613	120.6	4.285	$13\frac{1}{16}$	$2\frac{3}{4}$
5	4.856	127.0	4.535	$13\frac{1}{2}$	$2\frac{1}{2}$
$5\frac{1}{4}$	5.098	133.3	4.790	$13\frac{1}{3}$	$2\frac{1}{2}$
$5\frac{1}{2}$	5.341	139.7	5.020	$14\frac{7}{16}$	$2\frac{1}{2}$
$5\frac{3}{4}$	5.584	146.0	5.238	$14\frac{1}{2}$	$2\frac{1}{2}$
6	5.827	152.4	5.488	15	$2\frac{1}{2}$

in America, are by $\frac{1}{16}$, by $\frac{1}{8}$, by $\frac{1}{4}$, etc. In Germany, similar bars advance in size by 1 millimeter, up to 40, by 2 millimeters, from 40 up to 80, and by 5 millimeters above 80. This system may be memorized by 40 and 80—by 1, 2 and 5. It is the best that can be done with a system tied up to an unhalvable scale. It does not agree with the British or American sizes except in a few sizes. The system of bolts, diameters and threads per inch,

TABLE OF SCREWS.—BY REULEAUX.

Diameter of screw. d = mm.	Diameter at root of thread. d = mm.	Threads per 10 mm.	Thickness of head. Millimeters.	Whitworth's.		
				Diameter of screws. d = Eng. in.	Threads per 1 in. Eng.	Threads per length = diam.
6	4.1	7	7	$\frac{1}{4}$	20	5
8	5.9	6	8	$\frac{5}{16}$	18	$5\frac{5}{8}$
10	7.7	$5\frac{1}{2}$	10	$\frac{3}{8}$	16	6
12	9.5	5	11	$\frac{7}{16}$	12	6
15	12.2	$4\frac{1}{2}$	13	$\frac{1}{2}$	11	$6\frac{7}{8}$
18	14.9	4	15	$\frac{9}{16}$	10	$7\frac{1}{2}$
21	17.6	$3\frac{3}{4}$	17	$\frac{5}{8}$	9	$7\frac{7}{8}$
24	20.3	3	20	1	8	8
27	23.0	3	22	$1\frac{1}{8}$	7	$7\frac{7}{8}$
30	25.7	$2\frac{1}{2}$	24	$1\frac{1}{4}$	7	$8\frac{3}{4}$
34	29.3	$2\frac{1}{2}$	27	$1\frac{3}{8}$	6	$8\frac{1}{4}$
38	32.9	$2\frac{1}{2}$	29	$1\frac{1}{2}$	6	9
42	36.5	$2\frac{1}{2}$	32	$1\frac{5}{8}$	5	$8\frac{1}{2}$
46	40.1	$2\frac{1}{2}$	35	$1\frac{3}{4}$	5	$8\frac{3}{4}$
50	43.1	$1\frac{7}{8}$	38	$1\frac{7}{8}$	$4\frac{1}{2}$	$8\frac{7}{16}$
55	48.2	$1\frac{7}{8}$	41	2	$4\frac{1}{2}$	9
60	52.7	$1\frac{7}{8}$	45	$2\frac{1}{4}$	4	9
65	57.2	$1\frac{7}{8}$	48	$2\frac{1}{2}$	4	10
70	61.7	$1\frac{7}{8}$	52	$2\frac{3}{4}$	$3\frac{1}{2}$	$9\frac{5}{8}$
75	66.2	$1\frac{7}{8}$	55	3	$3\frac{1}{2}$	$10\frac{1}{2}$

common or general over the continent of Europe, is that known as the Whitworth system. They still adhere to this system, as they do to the British system of gas and steam pipes and their fittings. The Whitworth system pitches its threads to even numbers or half numbers per inch in length. These pitches are easily obtained from the lead screws of all lathes, which are 2, 4 or 6 threads per inch, as a rule. Having given up the inch, the Germans formulate their threads per diameter (see tables at the end of this paper). For the names of the bolts, they must either retain their English names, and call a 25.4-millimeter bolt one inch, or they must call it what it is, 25.4 millimeters, but some call it 25 millimeters, and make it .4 millimeters larger. This inch bolt has eight threads per inch, and, as the diameter, too, is one inch, it can be said to have eight threads per diameter.

* * * * *

Our metrological reformers urge us to adopt a new system in place of our present one, a system that harmonizes in no way with anything we now use. This new system is practically based on a certain measure over 39 inches long. This is cut up into 1,000 parts, and 100 of these parts cubed give their primary vessel of measurement. The contents of this vessel in distilled water under certain conditions form their pound weight. Had the British yard of 36 inches been so treated it would have been as good a system, but no better. It would have been as inapplicable comfortably to our profession as is the metric. The wonderful extension of the metric system to time and infinite

TABLE OF AMERICAN AND BRITISH GAS PIPES.

Nominal diameter.	U. S. standard.		British standard.		
	Act'l outs. diam. in Engl. inch.	No. of threads per inch.	Act'l outs. diam. in Engl. inch.	Act'l outs. diam. in mm.	No. of threads per inch.
⅛	0.405	27	0.406	10.318	19
¼	0.54	18	0.531	13.493	19
⅜	0.675	18	0.625	15.875	19
½	0.84	14	0.812	20.637	14
⅝	.....	...	0.906	23.018	14
¾	1.05	14	1.031	26.194	14
⅞	.....	...	1.187	30.162	11
1	1.315	11½	1.312	33.337	11
1¼	1.66	11½	1.625	41.274	11
1½	1.9	11½	1.875	47.624	11
1¾	.....	...	2.125	53.974	11
2	2.375	11½	2.375	60.325	11
2¼	.....	...	2.625	66.674	11
2½	2.875	8	3.0	76.199	11
2¾	.....	...	3.125	79.374	11
3	3.5	8	3.5	88.898	11
3½	4.0	8	3.937	100.01	11
4	4.5	8	4.437	112.71	11
4½	5.0	8	.....	.....	...
5	5.563	8	.....	.....	...
6	6.625	8	.....	.....	...
7	7.625	8	.....	.....	...
8	8.625	8	.....	.....	...
9	9.688	8	.....	.....	...
10	10.75	8	.....	.....	...

space was given up as impracticable long ago, and we are now asked to bear the shock of a mighty change to use this inconvenient system, this unhandy system of ten, for the sake of uniformity with some other peoples of the world.

THE PROTECTIVE POWER OF PAINTS.

In a paper recently read before the Newcastle section of the Society of Chemical Industry, Mr. Harry Smith, F. I. C., describes a series of very interesting experiments upon the comparative protective powers of different paints as applied to iron work. Three series of experiments were made. In the first series a method originating with Mr. Max Toltze was employed; a number of iron dishes, 5 inches across and about ½ inch deep, were cleaned and carefully painted with two coats of the paint to be tested. These dishes were then filled with water, which was allowed to completely evaporate at the ordinary temperature of the laboratory, after which the dish was again filled up; this operation being repeated six times in the course of the six months over which the experiment extended. The paints used were prepared by grinding the pigments with linseed oil on granite rollers to a stiff paste, which was then thinned with best quality boiled linseed oil—itself capable of drying in seven hours to a hard film when painted on to a glass plate. Thus tested, the only paints which remained practically unaffected were red-lead or orange-lead paints, some of which, however, such as the "vermilionette" and the scarlet-red paints, contained also a certain proportion of aniline colors; whilst two of the red-lead paints contained, in the one case 45 per cent., and in the other 66 per cent., of barytes. All the other dishes were more or less rusted, the order of merit of the better paints being as follows: 1. Zinc white. 2. Equal parts zinc white and barytes. 3. Zinc white 3 parts, barytes 7 parts. 4. Lithopone (a mixture of zinc sulphide, zinc oxide and barium sulphate). 5. Pure white lead. 6. White lead, 5.37 parts, barytes 4.03 parts. 7. White lead 5.05 parts, barytes 4.21 parts. All the other paints, 36 in number, proved very inefficient, the first dish to show signs of rust being that painted simply with linseed oil.

In the second series of experiments a number of painted iron plates were exposed to the weather for a twelvemonth, and with the single exception of the plate painted simply with linseed oil, all withstood the test remarkably well. In the third series of tests, strips of iron were painted, and when the second coat was quite dry these strips were placed in wide-mouthed glass bottles, which were then nearly filled with water, and allowed to stand. The bottles were not closed, but the contents were protected from dirt by standing them under a shelf, there being about ½-inch of space between the top of the bottles and the underside of this shelf. The bottles were left untouched for three months. Some of the plates were sensibly affected within seven days; but those which successfully withstood the shallow-dish test also resisted this one most successfully. The fact that paints containing such large proportions of barytes, as some of the lead paints noted above, gave such excellent results, is of much interest, as it goes to show that this material can hardly be considered as a mere adulterant. In fact, one paint made up only of barytes and linseed oil give better results than an oxide-of-iron paint.

Mr. Smith refers with approval to the methods adopted in painting the Forth Bridge. All plates and bars for that structure were cleaned with steel scrapers and wire brushes, and then coated with hot linseed oil. As soon as possible after erection they received two coats of red-lead paint, which were subsequently followed by two coats of iron oxide. The life of the paint on the upper portions exposed to the weather is found to be about three years; but it must be added that the paint is then still in good condition, and on less important bridges would by many engineers be allowed to pass for several years longer. Experience shows, however, that in such cases the metal is liable to be deeply pitted. The paint inside the tubes of the Forth Bridge is as perfect as when first applied twelve years ago. The parts of the bridge most subject to rust are near the water, where the underside of the girders get sprayed with salt. The rusting commences on the rivetheads and the edges of the plates. These portions of the bridge are cleaned and repainted every year. The work is in

charge of Mr. A. Hunter, who has supplied Mr. Smith with the above particulars.—"Engineering."

THE ECONOMY OF MECHANICAL DRAFT.

The advantages of forced draft in steam generation have generally been supposed to rest in the better control which is given over the combustion, and the ability to meet sudden demands for steam promptly.

In a paper recently presented before the Boston Society of Civil Engineers by Mr. Walter B. Snow, and published in the "Journal of the Association of Engineering Societies," the question of the influence of mechanical draft upon the ultimate efficiency of steam boilers was discussed, and some points of interest upon that side of the subject were brought out.

After describing the various methods of producing mechanical draft by the plenum and vacuum systems, with details of various installations, Mr. Snow proceeded to discuss the several elements of efficiency. Taking a case of a plant of 2,400 horsepower of modern water-tube boilers, equipped with economizers and set with chimney draft and also with mechanical draft, all other things being assumed equal, it appears that there is a saving in first cost of $6,400 on a total cost of $79,100, by the use of the mechanical draft instead of the chimney. Again the employment of mechanical draft will permit the use of deeper fires and a higher rate of combustion, so that the full actual capacity will be obtained with boilers of but 2,000 horsepower capacity under the lighter draft, and with equal economy, so that a further saving of one-sixth of the first cost of boilers may be made.

Turning now to operative costs, it appears that with mechanical draft properly applied, a large portion of the heat which escapes up the chimney with natural draft may be saved. The actual power required to operate the fan is only about one-seventy-fifth that required to produce the same draft in a chimney by the ascending force of a column of heated air. The remainder of the heat may therefore be saved by the use of

suitable heat-abstracting devices, and this is the real source of the economy of mechanical draft.

In other words, by the use of the economizer for heating the feed water, and of the air heater for heating the air supplied to the furnace, the waste of the flue gases may be reduced to only five per cent. of the total heat value of the fuel. The economy due to better combustion caused by the higher furnace temperature is also a feature of the use of mechanical draft as well as the closer regulation of the air supply to the actual demands of the combustion. When to this is added the ability to burn low-grade fuels at good rates of combustion it will be seen that the opportunities for economy are great.

It is evident from the points which have thus been made, that the advantages of mechanical draft are not so much those which are inherent in the system as those which may be added to it or introduced because of it. Theoretically the furnace has no way of knowing how the air which it receives is brought to it. Practically it is possible so to bring the air that it may be better applied, and better controlled, and that less heat need be wasted, and more heat may be applied to useful work. These things the mechanical draft cannot do of itself, but it can render it possible for them to be done. Like many other good things, it requires to be set to work with judgment and kept at work with intelligence, and under such conditions it will do good work.—"Engineering Magazine."

GAS AS FUEL.

The advantages of gaseous fuel have frequently been set forth, and in respect of cleanliness, convenience and freedom from smoke it leaves nothing to be desired. The question of the economy of converting coal to gas in a producer and using the producer gas as fuel, should, however, be determined by a comparison of the results which would be obtained if the same fuel had been burned directly in a furnace. The method of making such a comparison is very clearly given in an article in a recent issue of the "Iron and Coal Trades Review," from which an abstract is here given.

In the conversion of coal to gas in a producer an indirect or partial burning takes place, and this burning has generally been considered to occasion a loss of about one-third of the heating power of the coal, this conclusion probably being based upon the fact that carbon burned to carbonic oxide parts with 30 per cent. of its heating power. While this is true as to the fixed carbon in the coal, it is not correct for the volatile carbon, which, so far from losing heating power, becomes rather the absorbent of much of the heating power the fixed carbon loses.

In order that a fair comparison may be made, the action of burning coal in an ordinary furnace should be considered.

"Before the first appearance of flame can kindle on a piece of *ordinary* coal, heat must have been absorbed and work done upon that piece. This work is the liberation of gas and vapor out of its solid mass. The change in volume which takes place on the gas and vapor substances so liberated is very great. A single cubic foot of coal may yield several hundreds of cubic feet of gas and vapor, and this too while the coal to outward appearance is suffering no diminution of its own bulk. Such a change in volume in any body has a close connection with the amount of heat which must disappear in working the change. Of the whole substance of the coal there is usually about one-third, in many cases a good deal more, over which there passes this great change in volume. From this it will be seen that heat absorption on a scale too great to be overlooked has to be reckoned with in the combustion of a lump of coal."

When the combustion takes place in a producer it is that of the fixed carbon of the coal, burning to carbonic oxide, the carbon being dissolved away from its solid form into a fluid, combustible gas, only the incombustible ashes remaining. From the heat produced by this burning is supplied whatever part may be necessary to drive out the vapor and gas. Whatever moisture may be present in the air supplied is decomposed, but the heat absorbed in this action is restored by its subsequent recomposition.

Having considered these actions, it is possible to make a comparison between the effect of the direct combustion of fuel with

that produced by its conversion to gas in a producer previous to combustion.

As an example, Lanarkshire splint coal is taken as a fair representation of the common manufacturing coals of Scotland, and for 100 units, burned in an open-fire grate, it is shown that there would result 548,733 calories, making deduction for the heat expended in expelling the gaseous matters. The same coal converted into gas in a producer, the gas being subsequently burned cold, would evolve 493,924 calories, thus showing a loss of but 9.98 per cent. by burning it as producer gas. Approximately similar results are obtained by taking an average of the coals of the whole United Kingdom.

When the coal contains very low fixed carbon and a corresponding high proportion of volatile carbon there would be no loss whatever in the conversion into gas, while for anthracite there would appear a loss as high as 25 per cent., or even more.

With such fuels as the shales the proportion of volatile carbon is so high that it would be entirely possible to mix a shale with anthracite in such proportion that no loss would appear by conversion of the mixed fuel to producer gas.

As it has been proposed to conduct the gasification at a sufficiently high temperature to fuse the ash to a slag, in order to prevent clinkering, and to facilitate the removal of the non-combustible, it is interesting to see that the computations show a loss of less than one-half of one per cent. of the full heating power of the coal for each five per cent. of earthy matter fused to liquid slag.

It appears then that in making producer gas with coal of average quality, the loss in heating power need not be much over 10 per cent., and that this 10 per cent. of loss may be wiped out by certain changes in the fuel.—"Engineering Magazine."

ONE AND TWO-WIRE SYSTEMS OF ELECTRIC DISTRIBUTION, COMPARATIVELY DISCUSSED.

The fact that the largest steamship in the world, the *Oceanic*, has recently been fitted with an electric plant for which the hull

serves as one set of main and branch conductors between the dynamos, lamps and motors, emphasizes the claims of advantage made for the one-wire system of ship lighting.

The usual American practice in ship plants is to run a complete copper circuit of two insulated wires, to which the dynamos, lamps and all current-consuming devices are connected. This system has no electrical connection with the hull of the ship; both terminals of the dynamos are carried to the switchboard and double-pole switches and fuses are provided for all circuits.

In the one-wire system of ship lighting, one terminal of each dynamo, usually the negative, is connected directly with the steel hull of the ship, commonly by a contact plate in which a cable from the dynamo terminates. The other, or positive dynamo terminal, is carried to the switchboard and connected through single-pole fuses and switches with each of the wires leaving it. The single wires going from the switchboard each represent one side of a circuit for which the hull of the ship is the other. Each wire connects with its group of lamps, motors or other devices, and the other terminal of each lamp or motor is secured to the ship's hull.

The advantages of the one-wire system include a saving in both materials and labor. As all of the switches are single pole, the weight of their metal parts is only about one-half of that for double-pole switches. Fuses are also being single pole, their parts require only about one-half the weight and space of the two-pole type. The smaller number of switch and fuse parts require a smaller and more simple switchboard for mounting them, also smaller distribution panels at other points. The greatest saving of materials affected by one-wire plants is in the weight of copper required for distribution.

The weight of wire required for the circuits of electric plants varies inversely as the electric pressure or volts consumed between their terminals, when the full rated current is flowing, other factors remaining constant. As two wires of equal size and weight connect each lamp or motor with the dynamos, the use of the ship's hull for one side of the circuit reduces the total weight of wire one-half, provided the sizes of wires used for the other side of the

circuit remain the same as in the two-wire system. The loss in a wiring system, however, is properly decided on questions of lamp regulation, heating of conductors and power loss, so that the same loss of pressure in the entire circuit is desirable whether the two-wire system or the one-wire with hull return be used. Hence it is said that, since the electrical resistance of the hull is very small, the resistance of the wires in the one-wire system may be twice as great as that of one-half of the circuit in the two-wire system; that is, may equal the entire resistance of the two-wire system. If this claim is correct, the wires of the one-wire system with hull return may be of only one half the size of those in an equivalent two-wire system, for the same loss of power and pressure, and the weight of copper necessary for the one-wire plan is, therefore, only one-quarter of that with two wires or a complete copper circuit.

Simple as it thus appears to save seventy-five per cent of the weight of copper necessary for two-wire ship circuits, there are very good reasons why no such saving can be made in practice, for the great majority of cases. A suitable loss of pressure in the wiring of ship plants usually ranges from two to five volts at full load, being largely decided by considerations of pressure regulation at lamps as well as some regard for loss of power. There is, however, another important factor that cannot be ignored, namely, the safe carrying capacity of the wires; that is, the rate at which current, as measured in ampères, may flow through them without causing so great a rise of temperature that they become a source of danger as to fire. Now the safe carrying capacity of a given wire is a constant quantity under the same conditions, depending upon its sectional area and the kind and quantity of its insulation, while the maximum loss of pressure that may occur in a conductor varies directly with its length, and may be very small if the conductor is quite short. In the design of the two-wire system for ship plants with losses of from two to five volts pressure in the wiring, the permitted pressure loss determines in most cases the size of conductors, rather than considerations of safe carrying capacity. The practical effect of the hull return is to cut the length of wires in two,

so that if the same loss is to be had in the wires of the one-wire system that would be allowed in the two-wire system for the same service, it will often be found that the permitted loss involves too great a rise of temperature in the wires, and is, therefore, larger than is permissible. In this case the only course is to select some larger wire whose safe carrying capacity will not be exceeded by the desired flow of ampères. In many cases, for the distances common in ship plants, this limit as to safe ampère capacity of wires will be found to require about the same sizes of conductors in many parts of the one-wire system as would be used for the two-wire.

Another practical limit to the reduction of the areas of wires in the one-wire plan is found in the fact that, for mechanical considerations as to strength, the smallest wire that should be used for general purposes in either the one or two-wire system is about .064 inch diameter, and much more of this particular size, as to length, will be used than of any other. The result of the considerations just named, together with the fact that the weight of insulation does not vary so fast as that of its contained copper wire, tends to keep the sizes of wires for the one-wire system and also their weight at about that in one-half of an equivalent two-wire circuit.

Even on the supposition that the hull of a ship offers no electrical resistance when used as the return for a wiring system, it is by no means possible, therefore, to reduce the section and weight of wire in the one-wire system to one-fourth of the amount required for the two-wire plan. A little consideration will show, however, that to regard the return circuit through the ship's hull as of no resistance is a mistake. One of the fixed rules for the erection of electric wires is that all joints between wires be soldered to insure a low resistance at these points. At switch, fuse and socket terminals clamp joints are permitted, but special contact parts of relatively large surface are provided at these places, and these clamping parts are always of brass or copper, the surface of which is much less subject to the formation of oxides than is that of iron or steel. Now the oxides that gradually form on the surfaces of metals are the main cause of

the increasing resistance of unsoldered joints. When metal surfaces are perfectly clean the resistance of a tight joint between them can be lowered but little by the use of solder, but the solder prevents the formation of oxides in the jointed surfaces. It is well known that any iron or steel surface to which damp air has access soon becomes coated with oxide, and this result can hardly be avoided in the hundreds and thousands of contacts between electrical fittings and the hull of a ship, when the latter is used for the return circuit. The result is that a resistance at these joints, which may be trifling when the fittings are first erected, is apt to become quite a serious matter at the end of five or ten years. The hull of a ship, made up with a great number of steel plates, obviously has many joints which an electric current flowing between distant points on the ship must cross. These joints between ship plates are, perhaps, less subject to corrosion than are those between electrical fittings and the hull, but there must be some action of this kind, and the scale on the plate when they are at first joined is rather a poor electrical conductor. Considering both the joints between electrical fixtures and the hull, and also those between the hull plates, it seems practically certain that whatever the resistance of a return circuit through the hull when the ship is new this resistance will gradually increase and ultimately reach a very material amount. The resistance of a well designed two-wire system for a ship plant is in any case small, and it is at least doubtful whether the ship's hull and the connections thereto would, after a term of years, have a less resistance than one side or half of a proper two-wire system. At a time when electric plants on board ship were quite small and performed no service apart from lighting, the hull was more apt to give satisfactory results as a return circuit than now, when many hundreds of ampères flow from one part of the ship to others for the operation of lamps, motors and heaters.

Considering the factors of mechanical strength, ampère carrying capacity, and the uncertain and increasing resistance of joints, it seems impracticable to reduce the wires of a one-wire system below the sizes that would be used in the corresponding half of a two-wire plant. If the same sizes of wire are used in the two

systems for a given case, the one-wire plant will require just one-half the weight of conductors necessary for the two-wire. Some labor will no doubt be saved in the one-wire plant, but its amount will not be so great as might be supposed, since twin-wire cable is not much harder to erect than a single wire, and the labor involved to secure proper attachments to the steel plates of the ship's hull cannot be neglected.

The cost of a wiring equipment on board ship must depend on a number of factors peculiar to each case, important among which are the loss of pressure permitted in conductors and the size of the ship. Allowing for the use of the same sizes of conductors in both systems, however, the total saving of labor and material by the one-wire plan will probably not exceed one dollar per lamp (16-candlepower) capacity of the plant for an average case. To offset this rather small saving of first cost for the one-wire plan, it has a few decided disadvantages. Whether electrolysis at the joints between the steel plates of a ship's hull will ever become of importance is as yet an open question, but a practical demonstration of such effects on the hull of a ship may prove an expensive lesson. That heavy electric currents can work great damage when passing out of iron surfaces where moisture is present, and that with little warning until destruction is nearly completed, now has been fully demonstrated to the dissatisfaction of water and gas companies in many parts of the country.

It sometimes happens that the terminals of a fuse become short circuited or joined through a burnout, or that an arc continues between fuse terminals after the fuse has been melted by a heavy short circuit at some point on the line which it protects. In a two-wire system either of these mishaps is usually cared for by the fuse on the other wire, since double-pole fuses are regularly used on two-wire work. In the one-wire plant there is no duplicate fuse to blow and open the connection, so a fuse designed to protect some larger wire must either blow out or else the wire which the faulty fuse connection was designed to protect must melt and thus produce a very real danger of fire. The known nature of an electric current is to expend the most of its energy

in the generation of heat at those points in its circuit where the resistance to its flow is the greatest. This is illustrated by the incandescent lamp, which consumes most of the energy of the current flowing through it simply because its resistance is usually many times that of the wiring to which it is attached. As the joints between the electrical fittings and the steel hull of the ship grow old some of them are apt to develop a high electrical resistance through being loose or rusted, and such loose joints may become a serious danger as fire risks, because of their development of heat, against which no fuse on the circuit can guard.

In a two-wire plant one-half of the total electric pressure acts, under normal conditions, to break down the insulation of either wire or pole, while in a one-wire system, with a hull return, the entire effort of the pressure is to pierce the insulation at the single-wire pole. For equal security against this effect alone, therefore, the insulation of a single-wire system should be of double thickness. It is regular practice in the erection of two-wire plants to install at the switchboard a device that will indicate at once when an accidental ground, or, in the case of ships, iron connection, occurs in any part of the electric circuit, so that it may be removed. As a one-wire circuit with hull return is in direct contact with the iron of the ship at a great number of points, an accidental contact in any place cannot be indicated at the switchboard, but is revealed only by the injurious effects it produces locally in the way of heat or fire at the place where it occurs. A very heavy iron connection might take so much current that the extra flow would be indicated by the ampèremeter, but this cannot be relied on, as the load of ampères is constantly changing in any event. On a two-wire system a single ground or iron connection produces no flow of current or bad results whatever until another ground or iron connection is made on the other side of the circuit, and there is usually opportunity to remove the first ground or iron connection before a second one can occur. With one wire and a hull return there is damage at once when the first iron connection occurs, and no chance to remedy the faulty contact until the trouble happens.

Under ordinary conditions a contact with terminals of about 110 volts pressure gives but a slight shock to most persons, but with 220-volt circuits, which are now coming into use for lamps and motors, an unpleasant sensation can be had. It is seldom that the user of lamps and fittings on an insulated two-wire system is exposed to a shock, as he must make contact with both wires at some uncovered points simultaneously, but with the one-wire plant a hand laid on an iron part of the ship, while the other is in contact with the wire or a switch will result in a discharge of current through the body.

In face of these facts the preference that has grown up for two-wire insulated circuits on ships seems to be well founded.—Alton D. Adams, in "Marine Engineering."

METALLIC BRUSHES FOR ROTARY CONVERTERS.

The carbon brush, which has become such a universal fixture of electric generators and motors, has also been generally used on rotary converters, probably from force of habit on the part of the designers rather than by virtue of any special fitness for this purpose. In generators and motors the high resistance of the carbon brush has been necessary in order to avoid sparking under load changes and corresponding shiftings of the neutral points. In the rotary converter the position of the neutral remains fixed, regardless of changes of load, so that the high resistance of the carbon brush is not needed in this machine to enforce sparkless commutation. As is well known, the benefits of the carbon brush are obtained at the cost of considerable loss. In addition to the resistance of the material of the brushes themselves, there is a contact resistance between the face of the brush and the surface of the commutator, which resistance has a peculiar property of varying almost exactly in inverse ratio with the current density, so that the drop across this contact is nearly uniform regardless of the current flow. This drop with the usual material and conditions is from $\frac{3}{4}$ of a volt to $1\frac{1}{4}$ volts, making a total drop across the contacts of both the positive and negative brushes varying from $1\frac{1}{2}$ to $2\frac{1}{2}$ volts.

A rotary also differs from a direct-current generator in having a much higher peripheral speed owing to the necessity that each part of the armature shall pass a large number of poles per second, and the difficulty of crowding the proper number of teeth and slots per pole in a very narrow space. Twenty-five-cycle rotaries have a peripheral speed of about 5,000 feet per minute and 60-cycle machines run considerably higher. In order to prevent the commutator segments from becoming too crowded and narrow, the commutators must have a diameter equal to about two-thirds of that of the armature, giving a peripheral commutator speed much higher than is customary in direct-current generators or motors. On this account the friction of the brushes becomes also a serious factor. The high speed also reduces the diameter and consequently the superficial area of the commutator from which the heating losses must be radiated, necessitating careful reduction of these losses to prevent overheating. The same high speed also involves another trouble in many cases where it becomes difficult to make the commutator segments of sufficient length to provide ample brush-bearing surface, the centrifugal force tending to buckle segments of too great a length or even to tear out their ends from clamping rings. The radial width of the brushes cannot, as a rule, be made as great as in generators, owing to the small polar pitch. For this reason, again, metal brushes are preferable to carbon brushes. On the other hand, accidents occasionally happen with rotaries which do not happen with generators, and which might be more disastrous with metal brushes than with those of carbon. Among such accidents is the dropping out of step, which causes severe arcing around the commutator, and which would do more damage to metal than to carbon brushes.

Rotary converters differ from direct-current generators in that their efficiencies are higher; hence the commutator losses are a much greater portion of the total waste of energy in the machine. For instance, large rotary converters commonly work at an efficiency of 95 or 96 per cent., their losses being little more than half of those of a mechanically-driven generator of the same size. With carbon brushes the losses of the commutator become a considerable part of the total losses of the machine. In a 125-

volt machine the drop due to the contact resistance between the brushes and the commutator face may alone amount to two per cent. of the total output, in addition to which there is chargeable against the commutator almost all of the friction loss of the machine. With rotaries delivering higher voltages, the contact drop becomes a smaller proportion of the total e. m. f., and hence the waste becomes less, but it still remains a large part of the energy dissipated in the machine. With copper brushes the fall of voltage across the contacts can easily be kept down to a much smaller value, such as one-tenth of one volt.—" Electrical World and Engineer."

THE DUTIES AND RESPONSIBILITIES OF THE NAVAL ENGINEER IN THE BRITISH NAVY.

Forty years ago the duties and responsibilities of the naval engineer officer were confined to the care, maintenance and manipulation of the engines, boilers, pumps, pipe connections, the sluice valves between the compartments, and all sea-inlet valves.

Some conception of the onerous duties and vast responsibilities which now devolve upon the chief engineer of a modern warship may be obtained from a perusal of the following extract from the Queen's regulations, stating the items of machinery and parts of the ship which are placed in his care, and for which he is held personally responsible.

1. The machinery and boilers of the ship and boats (the *Terrible* and *Powerful* have 48 boilers and 25,000 indicated horsepower engines, and cruisers are now building with 30,000 indicated horsepower engines; many ships have four steamboats).

2. All auxiliary machinery, for whatever purpose fitted.

3. All pumps, with the pipes, cocks and valves belonging to them.

4. All distilling apparatus, etc.

5. All gun mountings and torpedo carriages.

6. Propeller lifting apparatus.

7. All steam and hydraulic, pumping, and other engines for loading and working the guns, for supplying ammunition, and for turning turrets, barbettes, platforms, etc.

8. All ventilating engines and gear.

9. Capstan engines, shafting and spindle of capstan and windlass, and steam-steering engines and steering gear as far as the rudder, with spare gear for the same.

10. Hydraulic jacks, with the exception of those in the gunner's charge.

11. Steam winches and gear for hoisting in torpedo and other boats.

12. All watertight doors and sluice valves, including horizontal trap-and-flap doors, as well as vertical hinged doors.

13. Steam fire engines, and all pipes, cocks and valves in connection with the fire main.

14. Instruments and gear for telegraphing signals in connection with the machinery.

15. Whitehead torpedoes, submerged discharge tubes, and gear for torpedoes.

16. All air-compressing machinery, reservoirs, separators and charging columns.

17. Electric-light engines and dynamos.

18. All flooding gear, including valves, cocks, pipes and other fittings.

19. Refrigerating machinery.

20. All such other parts of the hull, double bottoms, and exposed iron surfaces as may be in his care, either wholly or jointly with other officers.

Items 5, 7 and 15, should be specially noted.

In addition to the above, the regulations also state: "That the chief and other engine-room artificers, and chief and other stokers, are to be under the immediate direction of the engineer of the respective watches, the engineer officer to be responsible for the general decorum, good order and cleanliness of the engine-room, and he will see that the engineers and the other persons employed under his control perform their duties with promptitude and to the best of their abilities." Duties of an executive nature connected with evolutions and inspections of the engine-room ratings and their clothing, also devolve upon him, and make an appreciable demand upon his time.—"Engineering."

OUR EXPORTS OF IRON AND STEEL.

No feature of the marvellous growth of our commerce is more striking than that relating to exports of iron and steel. The total foreign commerce of the United States in the year just ended has for the first time crossed the $2,000,000,000 line, and the total exports of the manufacturers of iron and steel have for the first time crossed the $100,000,000 line. In the calendar year 1890, the total exports of iron and steel amounted to only $27,000,000, but in 1899 they were $105,689,645. In the same period the importations of manufactures of iron and steel have decreased with nearly equal rapidity, the importations of 1890 being $44,544,140, while those of 1899 were $15,799,206. The striking feature of this rapid growth in our exportation of manufactures of iron and steel is the fact that European countries are taking largely from us in these lines.

In builders' hardware, for instance, the United Kingdom took nearly $2,000,000 worth in the year just ended, and Germany more than $1,000,000 worth; and the exports to the United Kingdom of sewing machines were $1,285,609 in 1899, against $806,401 in the preceding year, and the trade in the same line with other countries was also gratifying. For new and ingenious machinery the world seems now to be looking to the United States. Exports of electrical machinery increased from $917,453 in 1897, to $2,523,644 in 1898, and $3,143,336 in 1899, and metal-working machinery from about $4,000,000 in 1897 to nearly $7,000,000 in 1899. Railway engines increased from $3,000,000 in 1897 to nearly $5,000,000 in 1899; typewriting machines from $1,566,916 in 1897 to $2,776,363 in 1899. Such lines of machinery as cash registers, laundry machinery, printing presses, shoe-manufacturing machinery, fire and stationary engines show a marked growth.

In 1880, the production of pig iron in tons in the United States was 3,835,191 tons. The value of iron and steel manufacturers exports was $15,422,874, while the imports amounted to $63,956,853 in manufactures of the same line. Nineteen years later, in the calendar year 1899, the pig-iron production amounted to 13,620,703 tons, while the exports of iron and steel manufac-

ture amounted to $105,689,645, while the imports in the same line shrunk to $15,790,206.—"Scientific American."

ELECTROLYTIC COPPER TUBES.

With the increase in steam pressures and consequent increase in temperatures the demand for seamless copper tubing for steam connections and similar purposes has led to various attempts to produce tubes of any required diameter without the necessity of using brazed seams. Among these it is but natural that the idea of employing electrically-deposited copper should have been considered, and so long ago as 1876 experiments were made in Russia, the land where the electro-deposition of metals originated, and tubing of moderate dimensions successfully produced. The principal difficulty in practice appears to have been the granular character of the deposit, thus causing its strength to be irregular, and frequently far below that of the rolled metal, and recent efforts have been mainly directed toward the development of some method by which pure fibrous copper of maximum strength may be deposited electrically even when a high-current density is employed.

Various plans have been tried to obtain a uniform and tough deposit of copper, and in a paper recently presented before the Institution of Electrical Engineers, Mr. Sherard Cowper-Coles reviews these, and describes the centrifugal method, devised by himself. The well-known Elmore process employs an agate burnisher for the purpose of consolidating the copper during deposition. When this method is used for making tubes the cathode on which the copper is deposited is slowly revolved while the burnisher traverses back and forth, producing a copper of high density, the specific gravity being as high as 9.2 and the strength 26.5 tons per square inch. The principal drawback to this process is the tendency of the copper to exfoliate under the burnisher, the pressure requiring to be regulated to a great nicety. Another method employs a sheepskin burnisher instead of a metal one, the effect being to cover all projecting parts with a thin film of animal fat, which hinders further deposition until the surround-

ing depressions have been raised to the common level. Various other forms of impregnators have been employed for a similar purpose. Other methods have been devised in which the electrolyte is injected upon the copper during deposition, but none of these has as yet come into extended use.

In the centrifugal method, devised by Mr. Cowper-Coles, no burnishing is used, but the mandrel is caused to revolve at a high rate of speed, while the copper is being deposited upon it. Several results are produced by this action. The skin friction between the deposited metal and the electrolyte keeps the copper smooth and compact, and at the same time any gas bubbles evolved at the cathode, or any impurities in mechanical suspension in the electrolyte, are thrown off by centrifugal force, and hence the principal sources of defects are removed. Porosity is largely due to the formation of gas bubbles, while excresences are mainly due to the presence of specks of foreign particles. In addition to the revolving motion which is given to the mandrel, the electrolyte is briskly circulated through the cells by an acid-proof pump, the solution being forced or pumped to a reservoir, where it is passed through a filter to rid it of all impurities in suspension.

By the employment of this method it has been found possible to produce smooth and tough deposits with the remarkably high-current density of 190 ampères per square foot, the voltage being about 1 to 1.2. It is also practicable to use a hot and more acid solution than has generally been employed, this allowing a heavier current to pass at a much lower voltage than would otherwise be possible, the deposit also being much tougher than when a cold solution is used. Some copper sheet, not annealed, deposited by the centrifugal process, 0.0245 inch in thickness, broke at 433 pounds, which is equal to 22.1 tons per square inch. Hard drawn wire made by this process has a tensile strength of 29 tons per square inch, the electrical conductivity being 99 per cent.

It is rather interesting to note that formerly copper pipes were made by bending and brazing rolled sheet copper, while at the present time it is found desirable to produce sheet copper by

electrolytically depositing a seamless tube and subsequently cutting it open and flattening it out.

The development of this branch of electrometallurgy is an excellent instance of the manner in which electrical processes are modifying and in some cases revolutionizing older methods. It is authoritatively stated that more than one-third of the copper produced in the world is now electrolytically refined, one-half of the output of the United States being so produced, and nearly all the copper used by electrical engineers is refined electrolytically.—"Engineering Magazine."

PROPER SIZES OF PIPES FOR HOT-AIR HOUSE HEATING.

In a series of articles on "Furnace Heating," by Mr. Wm. G. Snow, published in "The Metal Worker," we find the accompanying table showing the sizes of air pipes for houses heated by hot-air furnaces. In connection with this the advice is given that the cold-air inlet to the furnace should be at least equal in cross-section to the combined cross-section of the pipes leading from the furnace. A branch from the cold-air duct should permit air to be drawn from the interior of the house in extremely cold or windy weather. The data upon which the table is based are set forth in Mr. Snow's paper as follows:

Much larger furnace pipes are now used than formerly. This involves a greater original outlay and an increased running expense for fuel, but the householder is repaid by the more healthful conditions secured through the supply of an ample volume of warm air in place of a small volume of intensely heated air. The pipes should be so proportioned that the several floors will be heated equally. The accompanying table will be found useful in determining their size. It must be borne in mind, however, that in heating and ventilating work no rule or table can be successfully used without a certain coefficient of common sense to allow for varying conditions.

The main steps involved in the calculation of this table are:

1. The determination of the loss of heat through the walls, windows and floor or ceiling of the room.

TABLE IV.—SHOWING THE PROPER SIZE OF FURNACE PIPES TO HEAT ROOMS OF VARIOUS DIMENSIONS WHEN TWO SIDES ARE EXPOSED.

Width of room.	Length of room 8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24	25	26	27	28	29	30
8	7	7	7	7	7	7	8	8	8	...	...	...	...	...	...	...	...	...	...	...	...	...	...
	8	8	8	8	8	8	9	9	9	...	...	...	...	...	...	...	...	...	...	...	...	...	...
9	...	7	7	7	7	8	8	8	8	8	8	...	...	...	...	...	...	...	...	...	...	...	...
	...	8	8	8	8	9	9	9	9	10	10	...	...	...	...	...	...	...	...	...	...	...	...
10	...	...	7	7	8	8	8	8	8	8	8	9	9	...	...	...	...	...	...	...	...	...	...
	...	...	8	8	9	9	9	9	10	10	10	10	10	...	...	...	...	...	...	...	...	...	...
11	...	...	...	8	8	8	8	8	8	8	9	9	9	9	9	...	...	...	...	...	...	...	...
	...	...	...	9	9	9	9	10	10	10	10	10	11	11	11	...	...	...	...	...	...	...	...
12	...	...	...	...	8	8	8	8	8	9	9	9	9	9	9	10	10	...	...	...	...	...	...
	...	...	...	...	9	9	10	10	10	10	10	11	11	11	11	12	12	...	...	...	...	...	...
13	...	...	...	...	...	8	8	8	9	9	9	9	9	9	10	10	10	10	10	...	...	...	...
	...	...	...	...	...	10	10	10	10	10	11	11	11	11	12	12	12	12	13	...	...	...	...
14	...	...	...	...	...	...	8	9	9	9	9	9	9	10	10	10	10	10	10	10	10	...	...
	...	...	...	...	...	...	10	10	10	11	11	11	11	12	12	12	12	13	13	13	13	...	...
15	...	...	...	...	...	...	...	9	9	9	9	9	10	10	10	10	10	10	10	10	11	11	11
	...	...	...	...	...	...	...	10	11	11	11	11	12	12	12	12	13	13	13	13	13	13	13
16	...	...	...	...	...	...	...	...	9	9	9	10	10	10	10	10	10	10	10	11	11	11	11
	...	...	...	...	...	...	...	...	11	11	11	12	12	12	12	13	13	13	13	13	13	13	13
17	...	...	...	...	...	...	...	...	...	9	10	10	10	10	10	10	10	10	11	11	11	11	12
	...	...	...	...	...	...	...	...	...	11	12	12	12	12	13	13	13	13	13	13	13	13	14
18	...	...	...	...	...	...	...	...	...	...	10	10	10	11	11	11	11	11	11	12	12	12	12
	...	...	...	...	...	...	...	...	...	...	12	12	12	13	13	13	13	13	13	14	14	14	14
19	...	...	...	...	...	...	...	...	...	...	...	10	11	11	11	11	11	11	12	12	12	12	12
	...	...	...	...	...	...	...	...	...	...	...	12	13	13	13	13	13	13	14	14	14	14	14
20	...	...	...	...	...	...	...	...	...	...	...	...	11	11	11	11	11	12	12	12	12	12	12
	...	...	...	...	...	...	...	...	...	...	...	...	13	13	13	13	13	14	14	14	14	14	14

Temperature at register, 140°; room, 70°; outside, 0°. Rooms 8 to 17 ft. in width assumed to be 9 ft. high; rooms 18 to 20 ft. in width assumed to be 10 ft. high. For other heights, temperatures or exposures make a suitable allowance. When first-floor pipes are longer than 15 ft., use one size larger than that stated.

One 12-in. pipe = two 9-in. pipes.
" 13-in. " = " 10-in. "
" 14-in. " = " 11-in. "
" 15-in. " = " 12-in. "
" 16-in. " = " 12-in. "
" 17-in. " = " 13-in. "

In the space opposite the numbers indicating the length and width of room, the lower number shows the size pipe for first floor, the upper number the size pipe for second floor.

For third floor use one size smaller than for second floor.

For rooms with three exposures increase pipe given in table in proportion to the exposure.

For halls use pipe of ample size to allow for loss of heat to second floor.

2. The volume of warm air required to offset this loss.

3. The velocity of air in the pipes.

The loss of heat is calculated by first reducing the total exposure to equivalent glass surface. This is done by adding to the actual glass surface one-quarter the area of exposed wood and plaster or brick walls and one-twentieth the area of floor or ceiling to cover the loss of heat to non-heated basement or attic. Ten per cent. is added where the exposure is severe to cover the increased loss of heat by transmission and by the leakage of air. The window* area assumed in calculating the table is one-fifth, or 20 per cent. of the entire exposure of the room. From the inspection of a number of plans this ratio was found to represent a liberal allowance for glass surface.

Having obtained the equivalent of glass surface (E. G. S.) multiply by 85 (the loss of heat per square foot of glass per hour with 70 degrees difference in temperature). The product will be the total loss of heat by transmission per hour.

The volume of warm air required to offset this loss depends on its temperature, which generally ranges from 120 degrees to 140 degrees in zero weather. Assuming the temperature of the entering air to be 140 degrees and that of the room to be 70 degrees, the air escaping at approximately the latter temperature will carry away one-half the heat brought in. The other half, corresponding to the drop in temperature from 140 to 70 degrees, is lost by transmission. With outside temperature zero, each cubic foot of air at 140 degrees brings into the room 2.2 heat units. Since only one-half of this, or 1.1 heat units, can be utilized to offset the loss by transmission, to ascertain the volume of air per hour at 140 degrees required to heat a given room, divide the loss of heat by transmission by 1.1; the quotient is the volume sought. This result divided by 60 gives the number of cubic feet per minute. Having determined the volume of air required per minute, if we know the velocity with which it will travel through the pipes, their area in square feet is readily

* Double windows, when tightly put in, transmit about three-fifths as much heat as a single window.

determined by dividing the volume by the velocity in feet per minute.

In calculating the table, maximum velocities of 280 and 400 feet were used for pipes leading to the first and second floors respectively. These velocities are readily attainable in practice. They are lower than those commonly assumed for straight vertical flues, but this is accounted for by the greater resistance to the passage of air through the nearly horizontal basement pipes, and through elbows, nettings and registers. The size of the smaller pipes was based on lower velocities, according to their size, to allow for their greater resistance and loss of temperature.—" Engineering News."

A QUESTION OF BILLETS OR TUBES.

The United States General Appraisers at New York have rendered the following decision relative to the duty on tubes drawn from hollow billets, recognized commercially as finished tubes:

The merchandise in question was returned by the local appraiser as " finished steel tubes, not specially provided for," and duty was assessed thereon at the rate of 35 per cent. ad valorem under the provisions of paragraph 152 of the act of July 24, 1897.

The importers claim that the merchandise is dutiable at the rate of $1\frac{2}{10}$ cents per pound as " hollow steel billets," under the provisions of paragraph 135 of said act, or as forms or shapes of steel not otherwise provided for.

Counsel for the importers claims that the precise question raised by these protests has been frequently passed upon by this board, in G. A. 1,453, G. A. 3,342, G. A. 3,359 and G. A. 3,526; that it is *res adjudicata*, and that those adjudications should be followed. As to the first three of those decisions, the articles there in question were 18 inches long and about $\frac{1}{2}$ inch in thickness of walls, while in the present case the sample introduced by the importers is about 6 feet 8 inches in length, and those taken from the importation by the examiner about 12 feet and 6 feet long, respectively, the thickness of the walls in each of the three samples being about $\frac{7}{40}$ inch. As to board decision

3,526, the merchandise passed upon in that case was found by the board, upon the evidence in the case, to be known as ingots, and commercially unsuitable for any use as tubes. From the testimony in the present case it appears that a hollow steel billet is a short, heavy piece of steel which has been pierced through its entire length, and that such articles are made for the purpose of drawing out into tubes of a required size. When so pierced the walls of a hollow billet are very thick, and when the billet is drawn these walls are naturally reduced to the size desired. After the pierced block or billet has been subjected to a process of drawing it is no longer a billet, but is a tube. Of such character is the article before us, and we find from the testimony in the case that it is commercially known and dealt in as a tube.

The evidence offered by the importers shows that these tubes were sold to the Pope Tube Company, who subjected them to further drawing, and thus made them into bicycle tubes, and the protestants contend that such use establishes their claim that they are billets and not tubes. The fact that the articles have been further advanced after importation to form a particular kind of tube cannot, however, operate to change their character as tubes if they were such when imported. These articles ceased to be billets and became tubes when they were drawn into their present condition.

The remaining question to be determined is whether the articles are finished tubes. The language of the paragraph under which they were assessed is as follows:

"Lap-welded, butt-welded, seamed or jointed iron or steel boiler tubes, pipes, flues or stays, . . . 2 cents per pound; . . . all other iron or steel tubes, finished, not specially provided for in this act, 35 per centum ad valorem."

The testimony upon the question of the commercial condition of these tubes is conflicting, but the weight of the evidence tends to prove that while as a matter of fact and in ordinary parlance their rough and ragged ends may make them unfinished articles, yet they are not so considered in trade and commerce; and we find that they are, on the contrary, known in trade and commerce as "finished tubes."

In the case of Hedden *vs.* Richard (149 U. S., 346), the Supreme Court held that commercial understanding of "finished" articles will prevail over the ordinary and common understanding. Justice Shiras, in the opinion rendered in that case, said:

"We are of the opinion that as the collector offered to prove that the word in question, 'finished,' had, at and prior to the passage of the act of 1883, a particular trade meaning the court shall have considered the trade meaning, if established, as applicable to the matter at issue, and should have submitted the case to the jury with instructions to render a verdict for the importers, if they found that the furniture was not finished, within the trade meaning of the term, and for the collector if they found the contrary."

Furthermore, it appears in this case that tubes with rough edges, and in exactly similar condition to those in question here, are sold to boiler makers and are generally called boiler tubes, and that it is the practice of the consumer to buy them in this shape and cut them up and fit them into the particular boiler he is building or repairing. If the views of the importers be correct, no boiler tube could be considered finished unless it were cut to the required size and made ready to be put into the boiler into which it is to be built.

This board, in G. A. 4,251, passed upon a somewhat similar question, and held that a tube imported for the purpose of cutting up for use in spinning frames was nevertheless a finished tube.

We find that the articles in question are drawn steel tubes, finished, and accordingly overrule the protests and affirm the decisions of the collector.—"The Iron Age."

TECHNICAL EDUCATION—AN ECONOMIC NECESSITY.*

I have chosen as my theme "The economic aspect of technical education" because I desire to maintain the following proposition, viz: the continued growth and prosperity of any nation depend

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* Extracts from an address before the Central Railroad Club by Professor V. M. Alderson, of the Armour Institute of Technology.

ultimately upon the technical education of its people. I do not claim that business success, industrial prominence or material wealth are the only desiderata for a nation or for an individual, but I do urge that material prosperity, widespread and universal, is an essential foundation for artistic, philosophic or literary prosperity. The man who does most towards increasing the general material prosperity of a country, who calls to the aid of mankind all the forces of nature, is the man of applied science—usually called an engineer.

The natural products of a country, as coal and iron, may be sufficient, for the Lord seems to have dealt more bountifully with some than with others; or, a nation may, by its own efforts, bring about success. How can this be done? I believe that the most important element in giving one nation industrial supremacy over another is the technical education of its people. Foreign agents, consular reports, natural inventive ability, keen business sense, all aid, but they are, individually and collectively, inferior to the power that comes from having the workman, the superintendent and the manager each carefully trained to do his own moiety in a superlative manner.

Let me give an illustration: In 1883, at Crefeld, Germany, there was founded, at a cost of $210,000, a school for the sole purpose of teaching weaving and dyeing and such parts of the allied arts and sciences as were useful in the weaving and dyeing industry. The school was founded not as the result of any demand on the part of the merchants or the people, but as the result of the far-sighted policy of German thinkers who recognized the need of technical skill in the weaving industry. The outcome was remarkable in the extreme, affecting not only the textile industry at Crefeld, but in all Germany, to such an extent that silk goods formerly made in France are now made at Crefeld and exported to France. The city dates its rejuvenation from 1883, when the textile school was established. Since that time the city has prospered marvelously. The school itself has served for a model for schools the world over. The possession of its certificate is a sure passport to employment, for the manufacturers in the textile industries seek out and employ the graduates, well knowing that

their own business success lies in having competent and thoroughly trained workmen.

And yet in Manchester, England, where the weaving and dyeing industries are of greatest importance, they are just beginning to establish a school of weaving and dyeing.

Let us now focus our attention and consider England, Germany and the United States, their relations to each other and their relative places in the contest for industrial supremacy. For three centuries England had been laboring industriously to acquire and to hold the commercial supremacy of the world. The extension of her empire has been along the lines of trade and traffic. She has used, as a means to this end, legislation to improve the condition of the workmen, subsidies, exclusive charters and privileges, early development of inventions and correct business principles. To a certain extent she has been successful—she has been mistress of the seas; she has monopolized the business of the world. But that a revolution has been in operation during the last quarter of a century no calm observer of the trend of events will deny. England has been steadily losing ground, and if she continues to lose her foreign trade as rapidly during the next quarter of a century as she has in the past, she will soon lose her commanding position. In 1870, Great Britain, exclusive of her colonies, did one-quarter of the world's commerce, and including her colonies, 35 per cent. In 1895 her share had fallen to 18 per cent., or, including her colonies, to 31 per cent., showing that while she still held the lion's share, that share was steadily diminishing. From another point of view: Between 1870 and 1895 British exports increased 13.17 per cent.; Russian, 17.35 per cent.; French, 20.40 per cent.; German, 42.90 per cent.; while the exports from the United States increased 110.66 per cent., showing that during the last quarter of a century England's commercial advance has been the least of all.

Here is a condition the explanation of which is interesting alike to the Englishman, the German and the American. The average Englishman is primarily a business man, a trader, a man of facts and experience; and he frequently assumes a state of inordinate

self-complacency. The average German is a man of details, but also of ideals. The average American is not only a man of facts and ideals, but an inventor as well. The downward tendency of English trade was first noticed by some far-sighted Englishmen, about 1881, who saw that some of the markets of the world, which her business men up to that time had controlled, were slipping away from her. In true English style a Parliamentary investigation was undertaken, which showed that two active competitors had appeared, Germany and the United States, and that a hitherto unrecognized factor had largely contributed to the success of Germany. To the Americans was given the credit of superior natural inventiveness, and to the Germans superior technical training. Strong efforts were then brought to bear upon the British Government to remedy the evil which would surely overtake British commercial interests if steps were not taken to develop a system of technical education. Parliament made some meager appropriations, but no great change came until 1890, when, by a peculiar political combination, the beer tax of 6d. per barrel, amounting to £3,750,000, was to be had for the asking. The friends of technical education were alert and succeeded in having the entire sum appropriated to technical schools. With this artificial aid they increased rapidly. In 1882 there were 1,402 schools giving technical instruction to 68,581 pupils; in 1895 there were 2,673 schools and 190,386 pupils.

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The city of Manchester, famous as a center for freedom of thought, and always alive to the interests of her people, has clearly perceived the need of technical education. Under the supervision of the efficient secretary, Mr. J. R. Reynolds, 4,000 students are now receiving technical instruction in one or more subjects. The work heretofore has been widely scattered, but on July 26, 1895, the erection of a new central building was begun. It covers a ground area of 6,578 square yards, and will cost, exclusive of ground or equipment, $650,000. The director and secretary of the work made extended examination of similar institutions in Germany, France, Canada and the United States before the plans were drawn. As at Birmingham, much of the

instruction is given at night, and includes, in addition to the well-known subjects common to all such schools, instruction in gas manufacture, paper manufacture, bleaching, dyeing and printing, cotton manufacture, weaving, brick work, flour manufacture and brewing. The curriculum of the school is intended primarily to aid the development of the industries in and about Manchester, because it is recognized that only by such means can the city maintain its supremacy in the struggle for industrial success.

The polytechnics of London cover essentially the same field as the municipal technical schools elsewhere. Their instruction is generally given in the evening, and covers a wide range of subjects. Individual polytechnics, however, make some feature more prominent than others. The Southwestern Polytechnic aims to specialize upon the higher branches of engineering work. Northampton Institute specializes on art. The most famous polytechnic in London is the Central Polytechnic, in Regent street, founded and supported by Mr. Quintin Hogg. Its origin was humble in the extreme, being in a secluded corner alley near the Strand. Here Mr. Quintin Hogg gathered his first group of the Ragged Club, and taught them to read, using the Bible as his text, and as his light the uncertain flicker of a candle in the neck of a beer bottle. From such an origin grew the present Central Polytechnic with its 16,000 members.

It is evident that England recognizes the reason for her decadence, and is supplying technical instruction to large numbers of students, but mainly through the inefficient means of evening schools. Such a plan may bring forth an occasional genius, but can never serve as a foundation for national technical training. It may make somewhat better artisans, but since it can command only the fading energy of a man who has already done a day's work, it cannot be relied upon to enable England to compete with other nations that have organized systems which demand the undivided attention of the workers. Inasmuch as England is without a general public school system, as we know it, her system of technical training is unorganized, un-coordinated and very largely what we call trade-school instruction. And further-

more, it has as its persistent enemy that English bugbear, the practical man without a trace of theory in his composition, who blocks the advance of technical training in every way he can.

There are Englishmen, however, who see the great problem and do not hesitate to speak. Lord Rosebery, in a speech at the opening of the Technical Institute at Epsom in 1896, said that Great Britain was falling behind other counties in commercial enterprize from want of adequate facilities for technical instruction. And Lord Rosebery was well informed. Sir William Bailey at the presentation of prizes at the Hindley Technical School (October, 1897,) said: "Those who know anything of the work done in other countries, those who have studied the progress and skill in the first principles of the arts of industry and in the scientific use of force by our competitors abroad, know very much that should discount our pride and make us feel anxious about our commercial future prosperity. Until recently technical education has been much ignored, if not entirely neglected, in this country." Sir Andrew Noble has said: "In the middle of this century English machinery was immeasurably superior to any other. To Great Britain's remaining content with this state of things, and to seriously neglected technical instruction, I attribute the very much greater comparative progress that Germany, the United States and Switzerland have made in the last fifty years." Whether the awakening has come soon enough, time alone can tell; but of the reality of the danger there is little doubt, and the efforts to combat it will surely be made with energy and ability.

All observers of German progress, regardless of nationality, unite in attributing her success to her magnificent system of education—technical, scientific and classical. Thoroughness and careful attention to details seem to be the watchwords. Each German state aims to bring its system up to the highest state of efficiency. The system of elementary schools, realschule, technische hochschule, gymnasium, monotechnic, polytechnic, university, is perfect in every detail. Every trade or craft is given its due share of attention, and its monotechnic schools are located at the convenient centers of each particular industry; but government control allows no duplication of effort nor fool-

ish rivalry. In Prussia there are 248 monotechnic schools, distributed among the various trades, thirteen of which schools are devoted to the textile industry. High-grade technological instruction is as carefully prepared for as the more elementary trade instruction; no part of technical education is neglected.

To show the thoroughness with which the Germans have carried out their idea, consider this: If a paper dealing with any particular phase of industry, say the textile industry, is published anywhere in the world, it is immediately translated into German and circulated throughout the textile schools of Prussia; and more, all necessary models, illustrations or lantern slides, which may serve to make the matter clearer to the student, are prepared and sent out with the paper. Their aim seems to be in this matter to know all that the remainder of the world knows and just a little more.

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Hamburg has risen from the sixth largest port in Europe to nearly the first. Almost every issue of a German trade paper notices the building of new factories, the additions to old ones, or the launching of new industrial enterprizes. German cottons are sold in Manchester, and German steel in Sheffield and Leeds. "Made in Germany" is a more common brand than "Made in Great Britain." When the English required that articles of German manufacture should be marked "Made in Germany," it was intended as a sign of inferiority—a mark of reproach. The very reverse has been true, and the Englishman saw his error when the *Kaiser Wilhelm der Grosse*, one of the finest steamships afloat, steamed into the harbor of Southampton with a huge placard hanging over her side on which was printed in mammoth letters, "Made in Germany." Truly, the Teuton has put his hands deep down into the pockets of the Briton. It does not take an observer very long to recognize the supreme reason for all this; that while England may excel in some quarters on account of natural or acquired advantages, and is to-day nominally in the lead, yet she is fast yielding to both Germany and the United States.

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As true Americans we should carefully examine the weak parts of our educational and industrial system and strive to strengthen them. We should not rest satisfied with the inventive genius of a Fuller, a Whitney, a Howe or an Edison, but should recognize that permanent industrial success will only come when every workman, every foreman and every superintendent is trained to the highest degree of efficiency.

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The manual-training idea, first suggested by Victor Della-Vos, Director of the Imperial Technical School of Moscow, in 1866, is being well cared for in the United States, and is being extended constantly. The work done at such schools as the Mechanics Arts High School in Boston, the Rendge Manual Training School at Cambridge, Mass., and the Woodward Manual Training School at St. Louis, compares favorably with work done anywhere. Also intermediate schools, such as Pratt Institute in Brooklyn, Drexel Institute in Philadelphia, and Lewis Institute in Chicago, are exploring a field hitherto untrodden and are profoundly influencing American education for good.

The highest grade of technical education—professional engineering—is well done at many universities like Pennsylvania, Columbia and Wisconsin, and at private institutions like Worcester Polytechnic and Stevens Institute of Technology, while the Massachusetts Institute of Technology and Sibley College of Cornell University may be ranked on a par with Charlottenburg, in Germany, and the Polytechnicum at Zurich, Switzerland.

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The advantage which we in the United States have is threefold—abundance of raw material and fuel near at hand, superior machinery and an intelligent people. If we could but add the special or technical training of our people, we could then feel confident that in all the elements which go towards assuring success in commercial warfare, we should be superior to any country in the world.—"American Machinist."

MAGNETIC APPLIANCES FOR SHOP AND SHIP.

The application of the electro-magnet in the form of magnetic chucks, especially for light work on either reciprocating or rotary machines in the machine shop, has already proved to be a success, and their use is becoming rapidly extended as their many features of special advantage are recognized. Adjustment of work on the machine with such appliances is almost instantaneous and the non-interference of clamps and dogs renders it possible to readily surface the object not only to the extreme edges but to also cut to extreme thinness if so desired.

In the present almost universal accessibility of electrical connection for ships, even where they have no generating plant of their own, no difficulty lies in the way of utilizing such appliances, as the chucks are in reality merely electro-magnets in box-form, completely enclosing an electric coil which is thoroughly insulated. While of course only iron and steel work can be carried on with this attachment, and while also only the direct current can be utilized for magnetizing it, there is a broad field of application and usefulness which is welcoming the device.

Lately we noted the advent of a magnetic clutch, in which magnetic attraction between friction surfaces is made sufficiently strong to transmit all the required power in lines of shafting. The working parts of the clutches are composed of metal having a high permeability, so arranged as to become magnetized upon the passage of direct current through the coils with which they are provided. The two parts of the clutch can be attracted together in this way with a pressure greater than that obtained in mechanical clutches, and it is only a question of making the clutches of sufficient diameter to enable them to transmit power in any desired amount. The space occupied upon the shaft is only about twice the diameter of the shaft, and by using a flange, forged solid on the end of the shaft, they can be made to occupy even less space when used as cut-off couplings. The fact of their having no projecting surface or parts to catch the air when in operation the windage resistance is negligible. The greatest advantage, however, of this form of clutch over others is that it is self-contained, the "action and reaction" being within

the clutch itself, and consequently there is no resulting end thrust upon the shaft bearings and no additional friction load due to the operation of the clutch. A recently designed clutch of this kind, intended to transmit 3,000 horsepower at 150 revolutions per minute, measured 100 inches in diameter and the amount of current required to properly energize it was claimed to be no more than would be used by about four 16-candle-power incandescent lamps.

Other magnetic devices, especially for lifting purposes, are in constant use in many iron-working establishments, but the application of the principle in connection with ship machinery is more novel.

One recently (1898) patented apparatus which appears to have points of attraction in both senses of the word, is the magnetic brake, invented by G. M. Guerrant, and intended to act as a *governor* for marine engines. The governor arms instead of acting to throttle the steam admission when the engine races (and which has always proved so unsatisfactory on ship board) are designed to produce, instead, a contact between a disc upon the shaft and large permanent or electro magnets, which instantly act as a brake until reduced speed permits the governor arms to fall and releases the contact. How much can be advanced by this apparatus toward securing a perfect governor for the engines of ships is not settled as far as we are informed, but the large cargo steamer, running frequently in ballast and with propeller half submerged, appears to offer all the difficulties which the most ambitious inventor of governors could ask to have placed before him in order that he might apply his method of overcoming them and convince the skeptical engineer.

MISCELLANEOUS.

A NEW SUBSTITUTE FOR CELLULOID.—Celluloid, which finds such numerous uses nowadays, is somewhat objectionable on account of its ready imflammability, and for other reasons. According to a note in "Cosmos" (Paris, March 31), a substitute called cellulithe is now made from paper pulp, that is not open to these objections. Says that paper:

"The numerous uses to which paper is already put are well known, whether under the ordinary form or in the state of papier-maché. Now a new substance is obtained from paper pulp, which can be used for the same purposes as celluloid. The invention of this curious material, which has been named 'cellulithe,' is based on observations made long since on certain modifications undergone by paper pulp when subjected to long continued beating. When it has been thus treated in the . . . mill whose office is to wash and defiberize the rags, a transparent and elastic fluid is obtained which hardens rapidly on drying and gives great strength to the paper; it is supposed that a colloidal, amorphous hydrate of cellulose is produced which separates from the cells of the pulp and acts as a glue. This is similar, as "La Chronique Industrielle" remarks, to the theory of the formation of so-called vegetable parchment or parchment-paper. By the action of the sulphuric acid, the cellulose is changed into amyloid, which with an excess of water gives a gelatinous precipitate that unites the fibers and finally forms a transparent sheet resembling parchment, except in suppleness.

"To prepare cellulithe, an exclusively mechanical process is used; that is, the pulp is beaten for an extremely long time. According to the particular kind of material employed, and also to the speed of rotation of the cylinder that does the work of the machine, the operation may last anywhere between 40 and 150 hours; it is prolonged until there is obtained a homogeneous mucilaginous liquid in which all trace of fiber has disappeared. This is called picturesquely 'milk of cellulose,' and its appearance perfectly justifies this name. If colored cellulithe is desired, colors are added at this stage of the process, and as in the state of extreme division to which the material has now been reduced, it contains much air, which might interfere with its smoothness, it is boiled to drive this air out. At the end of two hours the 'milk of cellulose,' boiled and filtered, is received in a perforated vessel, and then the water that it still contains is evaporated, either in the open air, or preferably in an oven at 40 degrees centigrade. Finally, a paste is obtained, which hardens slowly and attains the consistency of horn, with a specific gravity of

4.5. Like horn, this cellulithe can be worked, and has the advantage of not being inflammable like celluloid. Before drying sawdust and lampblack may be added, and then the compound becomes similar to ebonite. There is no need to say that this new substance may be applied to divers uses, and that its cost is reasonable, by reason of the material of which it is made and also because of the ease of manufacture and working."—Translation made for "The Literary Digest."

RIGHTS OF SHOP INVENTORS IN GERMANY.—In our present issue Mr. Tecumseh Swift discusses "The Rights of Shop Inventors," and in connection with his subject the following report of a case in Germany, furnished by Consul General Richard Guenther, at Frankfort, will be read with interest:

"It appears that a foreman in the employ of a certain establishment invented a substance which his employer used in finishing "rustling" velvet. The use of this substance was considered a trade or business secret of the firm. The foreman, however, imparted the composition to others, and the firm had him tried under the law of May 27, 1898, entitled 'An act for the suppression of base competition' (Gesetz zur Bekaempfung des unlauteren Wettbetriebs). The foreman was found guilty in the lower court.

"The case was appealed to the Federal court and the defense made the point that the foreman had only imparted his own invention to others; that the same was his intellectual property. The Federal court dismissed the appeal on the following grounds:

"The foreman was employed as such by the firm, und therefore it was his duty to try experiments by which the methods of work could be improved.

"His achievement in inventing the substance was therefore only a part of the services which he owed to his employers. The invention belonged to his employers, and therefore the action of the foreman in imparting the secret to outsiders was in violation of the law referred to. Only an employé is instrusted with experiments which may lead to such inventions. A stranger, whose labor does not belong to the establishment, is excluded from such experiments. The foreman was admitted to the experiments only

on account of his employment by the firm, and it followed that the invention in question was made by him in consequence of his employment by the firm, for which he was paid."—"American Machinist."

NICKEL-STEEL RAILWAY RAILS have been rolled for the Pennsylvania R. R. Co., and laid on the west track of the famous Horse Shoe Curve near Altoona, Pa. A letter from Mr. Wm. H. Brown, M. Am. Soc. C. E., the Chief Engineer of the company, recently published in the "Railroad Gazette," gives the following interesting details of this installation: The order for 300 tons of 3 per cent. nickel-steel rails was placed with the Carnegie Steel Co., on June 26, 1899. The steel was made by the Bessemer process and the rail was rolled to the 100-pound Am. Soc. C. E. section. In rolling the nickel content caused "red shortness" to such an extent that the rolling resulted in only 220 tons of No. 1 and 57 tons of No. 2 rails, and 19 tons of the latter had to be thrown away because of piping. The average analysis was as follows: Carbon, 0.504 per cent.; phosphorus, 0.094 per cent.; manganese, 1 per cent., and nickel, 3.22 per cent. Under the straightening presses the rails showed great rigidity; twice the force ordinarily used being required to accomplish the cold straightening, and often the rail would spring back to its former position after being struck, showing no effect of the blow. In drilling, the hardness was even more marked, in some cases five twist drills of ordinary tool steel being used up in drilling one hole. After experimenting with different materials for drills, it was found that the best results were obtained by using Mushet steel drills without lubrication. The rails have not been in service long enough to afford any records of their wearing qualities.—"Engineering News."

NICKEL-BRONZE FOR HIGH-PRESSURE STEAM FITTINGS.—In a note recently contributed to the "Chemical News," Mr. Sergius Kern, of St. Petersburg, gives particulars of a certain nickel-bronze alloys which are being tried for high-pressure steam fittings in the Russian Navy. The alloys, he states, are but little subject to corrosion, and are in some respects preferable to steel

castings. One of the alloys tried had the following composition: Copper, 10 per cent.; nickel, 17.5 per cent.; zinc., 12.5 per cent. In the cast state this alloy proved to have a tensile strength of about 26 tons per square inch, combined with an elongation amounting to 23 per cent. in two inches. The fracture was fibrous, and the specimen would stand bending through an angle of 65 degrees round a radius of 1⅛ inches. Another alloy, which has also given good results, is made up in the following proportions: Copper, 70 per cent.; nickel, 20 per cent.; zinc., 10 per cent. The tensile strength in this instance was 36 tons per square inch, and the elongation 14 to 17 per cent. on a length of two inches. The specimen could be bent over a radius of 1⅛ inches through an angle of 30 degrees to 40 degrees. Under a drop test specimens 30 millimeters square in section were bent through a right angle without fracture. The alloys were made in an ordinary copper-melting crucible, capable of holding 150 pounds of metal. Half the copper and all the zinc were first melted, cast into ingots, and sheared into small pieces, and the preparation of the alloy completed in the following manner: Nickel was placed at the bottom of the crucible, and then a layer of the above copper-zinc alloy, followed by a layer of copper, the above alternation of strata being repeated till the full charge had been added to the crucible. A final layer of charcoal was then added. About 15 hours were needed to melt the charge, which was run into flat open moulds, and remelted twice before using for castings. About 0.75 per cent. of zinc was lost at each melt and had to be made good before pouring.

The alloys shrunk considerably on solidifying and large risers were necessary.—"Engineering."

THE FORGING PRESS.—In a paper recently published in the "Journal of the Franklin Institute" for January, Dr. Coleman Sellers remarks that the introduction of the Whitworth forging press marked an important advance in the working of iron and steel as compared with the costly steam hammers previously in use, and hydraulic presses became absolutely essential in perfecting the link-and-pin system of bridge construction. Some of the

forgings required for the 5,000 horsepower dynamos needed by the Niagara Falls Power Company could not, he stated, have been executed by means of any existing steam hammer in America or elsewhere, and the Bethlehem Iron Company were the first to introduce into the States the hydraulic system of forging on a large scale. Forging by pressure in place of impact by hammers enabled deformation of metal to be effected with the least expenditure of power and with great exactness, as was soon manifested in the readiness with which hollow shafts were produced. About 1893 the work at Niagara Falls called for steel rings of absolute uniform density, having an outside diameter of 11 feet 4 inches, with a width on the face of approximately 50 inches, and a thickness of over 5 inches, which necessitated the use of a press of greater capacity than any heretofore erected. The press at Bethlehem, combined with the Whitworth system of compressed steel ingots, was taxed to its utmost to make what was needed in this case. It is interesting to note, he added, that this work was the first product of machinery introduced mainly to furnish armor plates and the massive steel forgings needed for modern ships of war.

Staybolts Breakage.—Writing to the American "Engineer and Railroad Journal," Mr. R. Atkinson, the mechanical superintendent of the Canadian Pacific Railway, states that he finds the shape of a locomotive firebox has much to do with the breakage of staybolts. Thus, a sharp "ogee," connecting the flat side of the firebox with the circular part, is particularly destructive; and it is common to find a whole row or two lengthways of the box broken off. The next most troublesome portions are the front upper corners, and then the back corners, after which come the top row across the back sheet. The use of thick plates for the outer firebox materially increases the number of failures, and for this reason the Canadian Pacific line have for years past never made these more than $\frac{7}{16}$ inch. The diameter of the bolt makes little difference, stays $1\frac{1}{4}$ inches and $1\frac{3}{8}$ inches in diameter failing just as frequently as $\frac{7}{8}$-inch or 1-inch bolts. On this point we may ourselves remark that the larger staybolts might be expected to fail the more frequently

of the two. Turning staybolts down between the plates does not, Mr. Atkinson states, give any benefit; and drilling the stays at the ends, if anything, seems to increase the breakage, whilst not proving effective in rendering the failure apparent by leakage. All staybolts, he observes, seem to fail by cracking across at top and bottom, near the outside sheet, leaving a strip at the center to break off last. The upper crack is usually deeper than the lower one, and in the upper corners of the side sheets is not quite horizontal.

PRODUCTION OF FINE COPPER FOR 1898.—Table giving the production in tons of 2,240 pounds of fine copper for 1898 in various countries. Compiled from H. R. Merton's "Principal Copper Supplies" for 1899 and Rothwell's "Mineral Industry for 1898."

United States,	237,173
Spain and Portugal,	53,225
Japan,	25,175
Chile,	24,850
Germany,	20,085
Australasia,	18,000
Mexico,	10,435
Canada,	8,040
South Africa,	7,060
Russia,	6,000
Norway and Sweden,	4,095
Italy,	3,435
Peru,	3,040
Bolivia,	2,050
Newfoundland,	2,100
Austria-Hungary,	1,540
United Kingdom,	640
Argentina,	125
Algeria,	50
Total,	427,118

—Fred'k H. Hatch in "Engineering Magazine."

THE NEW ELEMENT, VICTORIUM.—Sir William Crookes has recently given an account to the Royal Society of his discovery of the new element which he calls victorium. It has a pale-brown color and dissolves easily in acids. Its oxide is less basic than that of yttrium, but more so than the greater part of the earths of the terbium group. The chemical properties of victorium differ in many respects from those of yttrium, but generally speaking it may be said to occupy an intermediate position beteen this element and terbium. It is admitted that the oxide of victorium has the formula $Ve^2 O^3$, its atomic weight is not far from 117. The photograph of the spectrum given by the oxide shows certain definite lines which have not been observed with any other body. The spectrum is obtained by the incandescence of the body in a vacuum tube; the light given off has been analyzed by a spectroscope of great precision and the exposure upon a photographic plate shows a series of interesting rays in the ultra-violet region. In order to examine the negative an apparatus has been constructed which will measure to the 1-100,000 inch.—"Scientific American."

COAL IN GERMANY.—The production of coal last year by the collieries comprised in the German coal syndicate was 48,024,014 tons. The corresponding output in 1898 was 44,865,536 tons, so that the production increased last year to the extent of 3,158,478 tons.

DOWN-DRAFT FORGES.—One of the most marked improvements in the equipment of blacksmith or anglesmith shops is the substitution of down-draft forges for the old fashioned fires with hood and vertical smoke pipe. The advantages of this modern system are many. No obstruction is offered to the handling of large or bulky material, and the use of overhead cranes is possible. There are no unsightly pipes to be taken care of, and as a consequence, the shop looks cleaner and can be better lighted—both items which contribute to the comfort of the men. Then the fouling of the air in the shops is avoided, as all smoke and gases are removed immediately on being generated at the fire, and carried through underground pipes to the point

of exit. The system adopted by the Buffalo Forge Company can be installed in two ways: First, with separate fans, that is a blower for furnishing the blast and an exhauster for carrying off the smoke and gases; second, with a combined blower and exhauster, in which a portion of air is forced through the blast piping for supplying air to the hearth, and the remainder discharged into the smoke pipe, thus creating a strong draft.—"Marine Engineering."

LARGE NEW DRYDOCK.—An extensive graving dock has been erected at Shanghai by the Shanghai Engineering, Shipbuilding and Docking Co. Vessels of 12,000 tons may be accommodated. The length on the blocks is 465 feet to the inner chase and 526 feet to the outer caisson, the total length on top being 571 feet. The width at entrance at the bottom is 64 feet and at the top 80 feet, while the width in the dock is 54 feet at bottom and 128 feet at the top, and the depth of water on the sill from 23 to 24 feet at ordinary spring tides. The caisson, which serves both chases, is of wood with trolley lines laid over the deck. The water in the dock can be pumped out in two and a-half hours by two centrifugal pumps of 22 and 27 inches diameter respectively, driven by an ordinary compound engine. A slipway to take a vessel of 350 tons weight, operated by a steam winch and intended also for hoisting at the sheerlegs, is being constructed.

A SALT-WATER DETECTOR between the condensers and boilers of cruisers has been determined upon by the British Admiralty, to obviate a reoccurrence of the breakdown of the *Pegasus*, last February. The condensers of this vessel leaked badly, and salt water in her boilers caused them to prime to such an extent that the cruiser was practically helpless for 16 hours off Cape Ushant. After temporary repairs to two boilers, she reached Plymouth. The detector referred to is practically an appliance for chemically testing the water during its passage between the condensers and boilers, at once indicating the admission of salt water.—"Engineering News."

THE DEEPEST OCEAN TEMPERATURES RECORDED were taken by the U. S. survey steamer *Nero* in sounding for the proposed submarine cable between Guam and Midway Islands. At a depth of 5,070 fathoms, or 30,420 feet, the temperature was 35.9 degrees Fahrenheit, and at 5,101 fathoms, or 30,606 feet, it was 36 degrees Fahrenheit. The previous deepest temperature record was taken by the British survey steamer *Penguin*, in depths of 4,700 and 4,800 fathoms.—"Engineering News."

A NAVAL BASE AND COALING STATION at Guam has been decided upon by the Navy Department. The complete survey of the harbor and part of the island has been received, and it is proposed to build a breakwater at Port San Louis d'Apra, to erect repair shops and to build a coal shed of 20,000 tons capacity, fitted with automatic loading machinery, etc. It has been found that a whole day can be saved on the voyage to Manila by running fast and recoaling at Guam. Guam occupies about the same relative position to the China Coast that Bermuda does to the Atlantic seaboard of the United States. It is 1,506 miles from Manila, 1,823 miles from Hong Kong, 1,700 miles from Shanghai, and 1,342 miles from Yokohama.—"Engineering News."

CARBORUNDUM.—In a paper published in "Stahl und Eisen," Mr. Fritz Lürman discusses carborundum as a substitute for ferro-silicon in steelmaking. The substance in question is, it will be remembered, a product of the electric furnace. Its composition averages 62 per cent. silicon, 35 per cent. carbon, 1.5 per cent. iron, and 1.5 per cent. aluminum, together with traces of calcium and magnesium. It is quite free from sulphur and phosphorus. It is moreover, an endothermic body, and yields a considerable amount of heat when decomposed. In use, 88 pounds of it are equivalent to 550 pounds of ferro-silicon, but at present prices it, the latter, is more economical, the carborundum costing in the neighborhood of £160 per ton.—"Engineering."

PARSONS' STEAM TURBINES.—The Westinghouse Electric and Manufacturing Company, who own the American patents of the

Parsons steam turbine, have recently put three of these to work. Each set consists of a 500-horsepower turbine coupled direct to a 300-kilowatt alternator, and these alternators are run in parallel. Exhaustive economy tests have been made of the sets with the following results:

	Steam per Electrical Horse-Power Hour. Pounds.
At full load,	16.4
At three-quarter load,	17.0
At one-half load,	18.2
At one-quarter load,	22.0

The boiler pressure was 125 pounds per square inch. The Westinghouse Company have now a turbo-generator of 1,500-kilowatt capacity in course of construction. The spindle of this machine, complete with its vanes, weighs 2,800 pounds, and is six feet in diameter across the largest set of turbines. The designed speed is 1,200 revolutions per minute. This generator is to be established at the works of the United Light and Power Company, New York.—"Engineering."

THE PRODUCTION OF ALUMINUM in the United States in 1899 was 6,500,000 pounds, which is an increase of 1,300,000 pounds over the output of 1898.

SHIPS.

UNITED STATES.

New Ships.—The Naval Appropriation Bill now before Congress contains the following provisions relative to the Increase of the Navy, and concerning which there will probably be no disagreement:

Increase of the Navy.—That for the purpose of further increasing the naval establishment of the United States, the President is hereby authorized to have constructed by contract two seagoing battleships, carrying the heaviest armor and most powerful ordnance for vessels of their class, upon a trial displacement of about thirteen thousand five hundred tons, and to have the highest practicable speed and great radius of action, and to cost, exclusive of armor and armament, not exceeding three million six hundred thousand dollars each; three armored cruisers of about thirteen thousand tons trial displacement, carrying the heaviest armor and most powerful ordnance for vessels of their class, and to have the highest practicable speed and great radius of action, and to cost, exclusive of armor and armament, not exceeding four million two hundred and fifty thousand dollars each; and three protected cruisers of about eight thousand tons trial displacement, carrying the most powerful ordnance for vessels of their class, and to have the highest speed compatible with good cruising qualities, and great radius of action, and to cost, exclusive of armament, not exceeding two million eight hundred thousand dollars each; and the contract for the construction of each of said vessels shall be awarded by the Secretary of the Navy to the lowest best responsible bidder, having in view the best results and most expeditious delivery, and not more than two of the vessels herein provided for shall be built in one yard or by one contracting party; and in the

construction of all said vessels all the provisions of the Act of March third, eighteen hundred and ninety-nine, entitled "An Act making appropriations for the naval service for the fiscal year ending June thirtieth, nineteen hundred, and for other purposes," shall be observed and followed; and subject to the provisions hereinafter made, two and not more than two of the aforesaid vessels shall be built on or near the coast of the Pacific Ocean, or in the waters connecting therewith: *Provided*, That if it shall appear to the satisfaction of the President from the biddings for such contracts, when the same are opened and examined by him, that said vessels, or any of them, can not be constructed on or near the coast of the Pacific Ocean at a cost not exceeding four per centum above the lowest accepted bid for the other vessels provided for in this Act, he shall authorize the construction of said vessels, or either of them, elsewhere in the United States, subject to the limitations as to cost hereinbefore provided.

Georgia, New Jersey, Pennsylvania.—The principal features of construction and equipment of these three coast-line battleships have finally been formulated and issued in the following circular preliminary to advertising.

1. In the act of Congress making appropriations for the naval service for the fiscal year year ending June 30, 1900, approved March 3, 1899, provision is made under "Increase of the Navy," for three sea-going coast-line battleships, carrying the heaviest armor and most powerful ordnance for vessels of their class, upon a trial displacement of about 13,500 tons; to have the highest practicable speed and great radius of action, and to cost, exclusive of armor and armament, not exceeding $3,600,000 each.

2. To carry out the provisions of the above-quoted act, the Department, in advertising for these vessels to be built by contract, will invite proposals as follows:

Class I. For the construction of hull and machinery of each of said vessels, including engines, boilers and their appurtenances and spare parts, and for their equipment complete in all respects except as hereinafter stated, and for the installation of ordnance and ordnance outfit, in accordance with

plans and specifications provided by the Secretary of the Navy.

Class II. For the construction of the hull and machinery of each of said vessels, including engines, boilers and their appurtenances and spare parts, and for their equipment complete in all respects except as hereinafter stated, and for the installation of ordnance and ordnance outfit, in accordance with plans and specifications submitted by the bidder.

3. In order that ample time may be given to all ship and marine engine builders who may desire to submit their own plans of the hull and machinery, or of either, this circular defining the general requirements of the Department for the vessels referred to, is issued.

4. Bidders may, if they so desire, adopt the plans and specifications of the Department for the hull and machinery, or any part of either, and embody them in their plans and specifications to be submitted with their proposals.

5. The specifications submitted by bidders under Class II must contain certain customary general requirements usually embodied in specifications prepared by the Department, and must be so drawn as to fully cover the detailed requirements of the Bureaus of Construction and Repair, Steam Engineering, Ordnance and Equipment of the Navy Department; for further information concerning which, application should be made to the respective bureaus concerned; and no proposal under this class will be considered unless accompanied by full and complete specifications of the hull and machinery, including engines, boilers and their appurtenances and spare parts, and such equipment as is supplied by the builder, with such calculation of weights and displacement as will demonstrate the satisfactory strength and stability of the whole structure.

6. Vessels under Class II must not exceed, on full load displacement, the maximum draught contemplated in the Department's design.

7. Copies of the Department's specifications will be furnished upon application to the Department, in which bidders under Class II must show by erasures and additions the exact nature

of the hull and machinery, including engines and boilers, and their appurtenances and spare parts, and equipment, complete in all respects, that they propose to build and furnish, with the weights thereof.

8. In considering bids submitted under Class I, the Department will, other things being equal, give due preference to such as guarantee the shortest time for completion; and in considering bids submitted under Class II will give due preference, other things being equal, to such designs as are considered best adapted for the United States naval service, and which offer the highest speed, the greatest coal capacity, the greatest amount of armored protection, the best distribution of armor, and the best disposition of battery, and to such bids as guarantee the shortest time for completion.

9. The Department reserves the right to change, within a period of six months after date of the contract, the number and caliber of the guns, the arrangement of the battery and turrets, including the adoption of the superposed turrets, and the thickness and distribution of the armor, provided such change shall not involve an increase of weights over and above those herein specified for the above mentioned purposes; such changes to be made without additional cost to the Government.

10. The Department will reserve the right to reject any and all bids not considered advantageous to the Government, and no bids will be considered which propose to furnish vessels of less than 13,500 tons trial displacement, or of less than 19 knots speed on trial, or having a bunker capacity of less than 1,900 tons of coal.

11. Trial displacement is to be understood as meaning the vessel and her equipment complete, with 900 tons of coal, plus two-thirds ammunition, plus two-thirds of total stores and provisions, crew, and 66 tons of feed water for steaming purposes.

12. Bids must state specifically the price for each vessel.

13. Each vessel is to be built, fitted and equipped complete for sea by the contractors, with the exception of sea stores, ordnance and ordnance outfit, and such other articles as are hereinafter enumerated as to be supplied by the Government.

14. The contractors will fit and secure in place all armor, ordnance and ordnance outfit.

15. Bidders under either class must state specifically the time for completion of each vessel included in their bids.

16. The maximum time allowed for completion will be limited by the Department to thirty-six months for each vessel, and no bids will be considered which propose to exceed that limit of time. Failure to complete the vessel or vessels within the time specified in the contracts will involve penalties as follows, namely, $300 per day for the first month succeeding the expiration of the period fixed by the contract and $600 per day thereafter until the vessel is completed and delivered as prescribed by the contract.

17. The speed trial of the proposed vessels will be conducted after the vessels are completed, in accordance with the terms of the contracts, subject to conditions prescribed by the Department, and if, on trial, the average speed shall equal or exceed a speed at sea of 19 knots per hour for four consecutive hours the vessel will be accepted, in so far as the speed is concerned.

18. If the speed falls below 19 knots and exceeds 18 knots per hour, the vessel will be accepted, as far as speed is concerned, at a reduced price, the reduction being at the rate of $50,000 per quarter knot deficiency of speed from 19 to 18½ knots, and at the rate of $100,000 per quarter knot deficiency of speed from 18½ to 18 knots.

19. If the speed falls below 18 knots per hour, the vessel will, in the discretion of the Secretary of the Navy be rejected, or accepted at a reduced price to be agreed upon between the Secretary and the contractors. In case of rejection any money that may have been paid the contractors on account shall be refunded.

20. The following elements and qualities are obtained and provided for in the Department's design, and their equivalent must be obtained and provided for in all designs submitted.

21. The vessels are to be driven by twin screws.

22. The hull is to be of steel, having a double bottom and close water-tight subdivisions.

23. The arrangement of decks above water is to be such as

will provide ample freeboard and good berthing accommodations, the after body of the vessel being carried to the same height as the forward body.

24. There will be two steel military masts, fitted with upper and lower fighting tops, and with search-light platforms, signal yards and gaffs.

25. It is contemplated to supply the following boats, namely:

1 50-foot steam cutter,
1 36-foot steam cutter,
2 36-foot launches,
4 30-foot cutters,
2 30-foot whaleboats,
1 30-foot gig whaleboat,
1 30-foot barge,
2 20-foot dinghies,
1 16-foot dinghy,
2 12-foot punts,
2 18-foot Carley life rafts,

all of which must be stowed so as to be as far as practicable free from injury from the blasts of the guns. Boat cranes with power to be supplied as required for handling the heavier boats, and such other davits, cradles, etc., to be furnished as are necessary for the proper and efficient handling, stowage and securing of all boats. Good facilities are to be provided for hoisting and lowering all boats with dispatch, and two whaleboats are to be so carried as to be rapidly lowered under all conditions of weather.

26. The armored protection to be as follows:

(a) A complete water-line belt 8 feet in width. This belt is to be of maximum thickness for a distance of about 190 feet abreast the engines and boilers, that is, 11 inches thick at its upper edge and maintaining this thickness downwards for a distance of 5 feet, from which point it tapers to 8 inches at its lower edge. This belt is to be completed to the bow and stern, as follows: For a distance of about 45 feet, forward and abaft the heavy belt above described, the plates are to be 9 inches in thickness at the upper edge, tapering uniformly to a thickness of 6 inches at the lower edge. The next course of plates, forward and aft for a distance

of about 20 feet, are to be 6 inches in thickness at the upper edge, tapering uniformly to be 4½ inches in thickness at the lower edge. The next course of plates, forward and aft for a distance of about 20 feet, to be 5 inches in thickness at the upper edge, tapering uniformly to a thickness of 4 inches at the lower edge. The belt is to be completed to the bow and stern with plates of uniform thickness of 4 inches.

(b) Above the main belt, and covering the central portion of the hull for a distance of about 245 feet, the ship's sides are to be protected by armor of a uniform thickness of 6 inches, extending up to the upper deck; this upper and lower casemate side armor being joined to the barbette of the after 12-inch turret by athwart-ship armor of 6 inches in thickness, and connected forward by inclined armor 6 inches in thickness, thus forming a central casement or redoubt which will contain twelve 6-inch guns.

(c) All 14-pounder guns are to be protected by 2-inch plates of a sufficient area to form efficient shields to the crews working them.

(d) The barbettes for the turrets for the 12-inch guns are to be 10 inches in thickness, except where they are enclosed in the casemate, where the thickness is reduced to 6 inches.

(e) The turrets for the 12-inch guns are to be 10 inches in thickness, except the port plates, which will be 11 inches in thickness. The tops of the 12-inch turrets will be 3 inches in thickness.

(f) The armor for the 8-inch turrets will be 6 inches thick, except the port plates, which will be 6½ inches in thickness. The barbettes for the 8-inch turrets will be 6 inches in thickness. The tops of the 8-inch turrets will be 1½ inches thick.

(g) The conning tower will be 9 inches thick, also its shield; its tube will be 6 inches in thickness and its floor 2 inches thick.

(h) The signal tower will be 5 inches in thickness.

27. The main armor belt is to be so placed that its upper edge will be at least 12 inches above water at deep-load draught.

28. The 6-inch guns on the gun deck are to be further pro-

tected by splinter bulkheads of 2½-inch nickel-steel, between each pair of guns.

29. Wherever armor backing is required, it will be of East India teak wood.

30. All the above-mentioned armor and the regular armor bolts therefor will be supplied by the Government, but the protective deck, all splinter bulkheads, armor shelves and armored decks, will be furnished by the contractor.

31. The estimated weight in detail of the above armor is as follows:

	Tons.
Main belt, about 190 feet long, 8 feet deep and 11 inches to 8 inches thick,	597
Taper belt, 9 inches to 4 inches,	407
Upper and lower casemate armor, 6 inches,	903
Forward incline and after athwartship vertical armor, 6 inches,	292
12-inch turrets, 10 inches, and 11-inch port plates,	293
12-inch barbettes, 10 inches and 6 inches,	454
8-inch turrets, 6 inches, and 6½-inch port plates,	183
8-inch barbettes, 6 inches, and tubes, 3 inches,	135
Conning tower, 9 inches, and tube, 6 inches,	63
Signal tower, 5 inches,	15
14-pounder protection, 2 inches,	67
Splinter bulkheads, 2½ inches,	52
Total armor,	3,461
Backing,	178
Bolts,	47
Total armor, bolts and backing,	3,686
Cellulose,	85
Total hull protection,	3,771

32. A curved protective deck is to extend throughout the vessel, 1½ inches thick on the flat and 3 inches thick on the slopes.

33. A cellulose belt, about 3 feet thick, of approved material,

is to be worked along the sides above the protective deck the entire length of the vessel.

34. No woodwork is to be incorporated into the hull or fittings, except such as is indispensable.

35. The main deck only is to be of wood, which is to be laid on a complete metal deck.

36. The lower decks will be covered with linoleum or other approved covering.

37. All wood work used in the construction of the hull, except the armor backing and battens for electric-light wires, is to be fireproofed by a process satisfactory to the Department.

38. The vessels are to be fitted with bilge keels.

39. The hull is to be subdivided by a suitable number of watertight bulkheads.

40. A complete system of drainage of all watertight compartments, double bottoms, etc., with manifold connections to the steam and hand pumps, is to be provided.

41. Special care is to be observed to provide for commodious and well ventilated engine and fire rooms, and the system of ventilation is to extend to all living spaces, coal bunkers, ammunition rooms, holds and store rooms, and to the turrets and gun casemates.

42. Wherever it may be deemed necessary by the Department to insulate any compartments or bulkheads, or parts thereof, to arrest or retard the transmission of heat, the contractors shall furnish an approved insulating material and fit the same for such purpose.

43. All magazines are to be insulated by properly ventilated air spaces of about four inches, separating them from adjoining enclosures, and they are to be fitted with proper arrangements for lighting, flooding and draining, and if they are so placed that they cannot be flooded direct from the sea, they are to be connected with the fire main and fitted with safety valves.

44. Arrangements shall be made for cooling the magazines, or certain specified ones, by means of compressed air, by connection with the refrigerating plant, or by other approved means, and the

magazines, or certain specified ones, shall be further insulated by an approved non-conductor of heat.

45. Scuttles shall be provided in the magazine doors, or in such ones as may be specified.

46. The coal bunkers are to be well arranged, with ample facilities for rapid coaling and for supplying the fire rooms with coal, and small subdivisions and inaccessible spaces are to be avoided as far as practicable.

47. Approved means shall be provided for opening and closing by power, as well as by hand, such watertight doors as may be designated, with suitable devices for indicating at the operating station whether they are opened or closed.

48. A fire main of approved material is to be provided and lead below the protective deck, with risers leading to the principal compartments, decks and platforms, each riser being fitted with a shut-off valve at its outlet and another near its junction with the main pipe, and such other valves as may be required to cut off certain sections of the main pipe.

49. Each vessel is to be fitted for a flag ship, and ample provision made for the accommodation of the full complement of officers and crew, which will comprise:

1 Flag officer,
1 Commanding officer,
1 Chief of staff,
20 Ward-room officers,
12 Junior officers,
10 Warrant officers,
658 Crew, including marines.

50. Commodious and well ventilated wash and bath rooms, with shower baths and with fresh and salt-water connections, are to be provided for officers and crew.

51. Suitable and well ventilated sick quarters, with bath and water closet, are to be provided.

52. Proper accommodations are to be provided for bags and hammocks; mess and clothes lockers; racks for ditty boxes; lockers for rain clothes, and lockers for life preservers sufficient for the crew.

53. Shot racks shall be provided as required.

54. Provision must be made by means of tanks, with the necessary pipe connections and pumps, for carrying not less than 10,000 gallons of fresh water for drinking and cooking purposes.

55. Spacious, accessible and well ventilated dynamo rooms of proper size to accommodate the electric plant, will be required.

56. The galley spaces to be commodious and well ventilated, and fitted with the necessary metal lockers, drawers, shelves, etc., with fresh and salt water and steam connections, hand pumps, etc.

57. The galleys to be fitted with ranges for the officers and crew, steam-jacketed kettles for the crew, a coffee urn for the crew, a bake oven, and all necessary galley appliances excepting cooking utensils, together with the spare parts usually supplied. There shall also be, in a separate enclosure, a range for the commander-in-chief.

58. A small and compact laundry, with capacity to wash for about seventy-five men, is to be provided and equipped with the following laundry machinery of approved type, viz: one washer, one hydro extractor, and one mangle; also with two soapstone set tubs, one soap tank, ironing table, and drying room. Line shafting, to be driven by electric motor, is to be supplied, and all necessary steam and water connections provided.

59. Provision must be made for chain lockers; lamp, and oil rooms, with copper tanks and brass drip pans; paint lockers; cold-storage rooms; pantries; flag lockers; dispensary; commissary store rooms; servants' wash room; band room; ship's library; drying rooms; firemen's wash room, with clothes' lockers; general store room; general work shop, with machine tools as hereinafter stated; prisons; sail rooms; bread rooms; ordnance workshop, with lockers, drawers and vise bench, and all other necessary and usual store rooms—all to be completely fitted up as required.

60. There shall be suitable separate cabins for the flag officer and for the commanding officer; mess rooms for the ward-room, junior, and warrant officers, and the following state-rooms will be

required, in addition to those in cabins, viz: for ward-room officers, 20; for junior officers, 12; for warrant officers, 10. Also one for chief of staff, with toilet room.

61. Provision will be made for carrying not less than three months' allowance of provisions and six months' allowance of clothing and small stores.

62. The following machine tools of approved make and design will be furnished and installed in the general workshop, with proper line and countershafting (the former to be run by an electric motor), and all necessary and usual spare parts and tools, viz:

One screw-cutting back-geared gap lathe, to swing 30 inches over the ways and to take 10 feet between centers.

One 14-inch screw-cutting back-geared lathe, to take 4 feet between centers.

One column-shaping machine, 26-inch stroke, 26-inch traverse.

One double-geared drilling machine, to drill up to $1\frac{1}{2}$ to 18 inches from edge of work.

One 16-inch sensitive drill.

One No. 1 universal milling machine.

One combined hand punch and shears to cut $\frac{3}{4}$-inch round iron and $\frac{5}{8}$-inch steel plate, with 6-inch shear blades, to punch $\frac{3}{8}$-inch holes in $\frac{5}{8}$-inch plate.

One emery grinder, two 12-inch by 2-inch wheels.

One 30-inch grindstone.

Six bench vises.

63. Ample facilities for convenient and rapid coaling must be provided by means of steam winches and derricks, and a sufficient number of well-placed chutes and coaling ports.

64. The following auxiliary machinery of approved make and design, in addition to that pertaining to the main engines and their dependencies, is to be supplied by the contractor, is to be operated by steam power, and is to be installed complete, with all the necessary steam and exhaust piping and fittings, viz:

Steam-steering engine.

Anchor engine and capstan.

Ash-hoisting engine in each fire room.

Dense air ice plant of approved type, with a cooling effect of three tons of ice per day.

Evaporating plant to consist of four units, each having a capacity of 4,750 gallons of fresh water per day; also a distilling apparatus capable of condensing at least 10,000 gallons per day.

Five deck winches of 30 horsepower each; each winch to have two hoisting drums, either of which may be operated independently of the other.

Engines of 50 horsepower for each of the heavy bolt cranes.

65. The following auxiliary machinery is to be supplied and installed by the contractor, and is to be operated by electric power:

Turret-turning gear.

Ammunition hoists, except those in turrets.

Blowers for hull and for turret ventilation.

66. The following auxiliary machinery will be supplied by the Government, is to be electrically driven, and is to be installed by the contractor:

Turret-ammunition hoists.

Rammers for heavy guns.

Elevating gear for heavy guns.

Air compressor for charging torpedo flasks.

67. The line shafting for the machinery in the general workshop will be electrically driven, the contractor supplying the motor therefor.

68. All electric motors and the controlling devices therefor will be supplied by the contractor, except for the torpedo air compressors.

69. Good and approved means of interior communication throughout the vessel will be required by means of telegraphs, telephones, electric bells, voice tubes, helm, shaft, battle and range transmitters and indicators, etc., with annunciators of approved type.

70. A commodious pilot house of bronze, well located and completely fitted with steam-steering wheel, engine-room telegraphs, indicators, voice tubes, drawers, lockers, transom, etc.,

is to be provided, also a berth for the commanding officer in an adjoining enclosure.

71. A chart house, with chart table, drawers, racks, lockers and transom is to be provided.

72. There will be such other stations for steering by hand and by steam as may be designated, including one steam-steering wheel on the bridge over the pilot house, with the necessary indicators and telegraphs.

73. There will be two flying bridges, one forward and one aft.

74. The conning tower will be provided with steam-steering wheel, engine-room telegraphs, and such means of communication with the important stations of the ship as shall be prescribed, and with the usual instruments for controlling the ship in action.

75. Platforms will be required for search lights as designated.

76. Suitably placed platforms for leadsmen must be provided on each side of the vessel.

77. Towing and mooring bitts, warping chocks, canopies, awning stanchions, coal derricks, etc., are to be provided in accordance with usual practice.

78. Provision is to be made for carrying on recessed bill-boards and for handling four anchors, though it is probable that but three will be carried, viz: two of Navy type and one patent stockless anchor; the necessary anchor davits, securing and tripping gear, controllers and compressors, are also to be provided; and the anchors should be so placed as to be as far as practicable clear of the blast of the forward turret guns.

79. The engines will be of the vertical, twin-screw, four-cylinder, triple-expansion type, of a combined I.H.P. of 19,000. The steam pressure will be 250 pounds. The stroke will be 4 feet. The ratio of H.P. to L.P. cylinder will be at least 1 to 7.20, and the diameters will be sufficient for the required I.H.P. at about 120 revolutions per minute. Each engine will be located in a separate watertight compartment. They will be provided with all the necessary auxiliaries and accessories in accordance with the latest practice of the Bureau of Steam Engineering.

80. There will be twenty-four boilers of the straight water-

tube type, placed in six watertight compartments. They will have at least 1,280 square feet of grate, and 56,500 square feet of heating surface, and must be able to furnish steam for the main engines and all the necessary auxiliary machinery with an average air pressure in the ash pits of not more than one inch of water. All the necessary auxiliaries and accessories will be provided for the efficient working of the boilers.

81. There will be three funnels, each 100 feet high above the base line.

82. Sixty-six tons of water will be carried in reserve tanks for use of the water-tube boilers.

83. The vessel is to be heated with steam throughout, and the usual steam-heating system will be provided and installed by the contractor.

84. The weight of all machinery and tools, stores and spare parts, will be about 1,730 tons. This weight must include all articles, irrespective of name or use, coming under the cognizance of the Bureau of Steam Engineering, including water in boilers, condensers, piping, etc., but excluding the reserve feed water in the double bottoms and tanks.

85. The armament will be as follows:

Main Battery: 4 12-inch B.L. rifles, 40 calibers in length.
8 8-inch B.L. rifles, 45 calibers in length.
12 6-inch B.L. rifles, 50 calibers in length.

Secondary Battery: 12 3-inch B.L. rifles (14-pounders).
12 3-pounders.
4 1-pounder automatic.
4 1-pounder single shot.
2 3-inch field guns.
2 Gatling guns.
6 caliber .30 automatic guns.

86. Suitable and well supported platforms must be provided for all the guns of the main and secondary batteries.

87. The above battery, complete, will be furnished by the Government, and will be mounted as follows:

(a) Four 12-inch guns in pairs, in two electrically-controlled,

balanced, elliptical turrets, having inclined portplates; such turrets being on the line of the keel, one forward and one aft, having an arc of fire of 270 degrees.

(b) Eight 8-inch guns in pairs, in four electrically-controlled, balanced turrets, one on each beam near the forward end of the superstructure and one on each beam near the after end of the superstructure, having an arc of fire from right ahead and astern to not less than 55 degrees forward and abaft the beam.

(c) On the main or gun deck twelve 6-inch guns to be mounted, in broadside, six on each side, having an arc of fire of 110 degrees; that is, 55 degrees forward of, and 55 degrees abaft the beam.

(d) The 14-pounders and 3-pounders are to be mounted in commanding positions, having as large an arc of fire as possible.

(e) Two automatic 1-pounders to be mounted in each of the lower military tops, and two single-shot 1-pounders to be mounted in each upper military top.

88. Arrangements must be made whereby all the guns which do not train fore-and-aft, or nearly so, can be quickly and conveniently dismounted, housed and secured, so as to leave the side practically clear; all dismounting gear to be supplied by the contractor.

89. All bolts for securing gun mounts, sockets, etc., will be furnished by the contractor, and the installation of all ordnance will be at his expense.

90. Should patterns of rail sockets be required, they will be supplied by the contractor.

91. The turret ammunition hoists will be supplied by the Government, but will be installed by the contractor.

92. There must be ample and special means for ventilating the turrets and gun casemates.

93. Suitable means must be provided for a rapid and efficient supply of ammunition for all guns of the main and secondary batteries.

94. The contractor will supply such trolleys and purchases as are necessary to transport ammunition from the ammunition rooms to the ammunition hoists.

95. The contractor shall furnish and install the necessary water system for washing out the bores of the turret guns.

96. The magazines to contain the ammunition should be of such capacity that about one-half the total supply may be stowed in each end of the vessel, with suitable working space in addition.

97. In addition to the turret ammunition hoists, there will be a sufficient of electric power hoists direct from the magazines to the battery decks to furnish an ample supply of ammunition to all the guns; also, suitable means for supplying the guns on the bridges and in the tops.

98. The rate at which ammunition should be supplied to the 12-inch turret hoists is one complete round of powder and projectile to each hoist every ninety seconds.

99. The rate at which ammunition should be supplied to the 8-inch turret hoists is one complete round of powder and projectile to each hoist every fifty seconds.

100. The rate of supply of ammunition for the 6-inch guns should be three complete rounds per minute to each gun.

101. The rate of supply of ammunition for the 14-pounder guns should be six rounds per minute, and the rate for the 3- and 1-pounder guns, ten rounds per minute.

102. Armories, suitably located and fitted with racks for the stowage of a full supply of rifles, revolvers and cutlasses, and provided with drawers, lockers and shelves for small articles, will be required.

103. Racks for the full allowance of rifles for the Marine Guard will be required, also hat and clothes lockers for the same.

104. Such battle order, range indicators and transmitters as may be furnished by the Government will be installed by the contractor, who will supply and install the electric conductors and other material necessary for such installation.

105. There will be two submerged torpedo tubes located in one compartment, as designated on the plan. These tubes, complete, with sluice valves, will be supplied by the Government, but will be installed by the contractor, who will also supply the hull castings therefor, to which the tubes attach. The torpedo

room will also be fitted with proper racks and cages for the stowage of six Whitehead 5-meter torpedoes, and convenient means are to be provided for the handling of such torpedoes by means of trolleys, etc. A magazine for the stowage of six warheads, to be known as the "torpedo-head room," will be fitted as near the torpedo-manipulating room as possible. A torpedo-firing station will be fitted vertically over each tube, or very nearly so, and will probably be on the deck above the torpedo room. Each station will be fitted with armored protection for the torpedo operator sufficient to shelter him from fire from 6-pounder and smaller guns, will have a proper port in the side of the ship for use of the torpedo director, and will have a firing circuit to the tube below, and appropriate means of communication with the tubes and with the conning tower.

106. The air compressors for charging the torpedoes and the accumulators and separators will be supplied by the Government, but will be installed by the contractor, who will also supply and install such piping as may be necessary, or, if electric power be used, the Government will supply the electric motors, controllers and compressors, and the contractor will install the same, and supply and install the necessary conductors.

107. The total weight of the armament and ordnance outfit, including the torpedo outfit, is 805 tons.

108. The total weight of the ammunition and stores is 612 tons.

109. The total ordnance weight is 1,417 tons.

110. Ordnance weights to be carried on trial displacement, namely, armament and two-thirds ammunition, 1,213 tons.

111. The following list gives the details of ordnance weights:

		Tons.
Armament:	4 12-inch guns and mounts,	331.6
	8 8-inch guns and mounts,	230.0
	12 6-inch guns, mounts and shields,	172.8
	12 14-pounder guns and mounts,	30.0
	12 3-pounder guns and mounts,	8.4
	8 1-pounder guns and mounts,	3.2
	2 3-inch field guns and mounts,	1.2

		Tons.
Armament:	2 Gatling guns and mounts,	1.2
	6 caliber .30 automatic guns, etc.,	0.8
	rifles and revolvers,	1.8
	torpedo outfit,	24.0
		805.0

		Tons.
Ammunition:	240 rounds 12-inch,	143.8
	1,000 rounds 8-inch,	180.8
	2,400 rounds 6 inch,	189.6
	3,000 rounds 14-pounder,	34.5
	6,000 rounds 3-pounder,	22.4
	All other ammunition,	20.9
	Stores and outfits,	20.0
		612.0
Total weight of armament and ammunition,		1,417.0

Ordnance weights on trial,

Armament,	805 tons.
Two-thirds ammunition and stores,	408 tons.
Total,	1,213 tons.

113. The electric generating plant will consist of ten units, each unit to consist of an engine and dynamo mounted on a combination bed plate. Two units shall have a rated output of 1,250 ampères at 80 volts, while the other eight units shall have a rated output of 625 ampères each at 80 volts. The total weight of the ten units complete must not exceed 166,000 pounds.

114. The total weight of the whole electric installation, including dynamos, engines, bed plates, all fittings, wiring, tools, stores, instruments, and six search lights must not exceed 152.4 tons.

115. The following other requirements, with regard to equipment, shall be observed:

(a) The ship is to be completely equipped by the contractor ready for sea, except that the following articles will be supplied by and at the expense of the Government, viz: Anchors, chains,

cables, hawsers, anchor buoys, grapnels, devil claws, mooring swivel, stationery, flag and signal apparatus except electric and oil lanterns, navigational instruments and implements, band instruments, library books, bags, hammocks, rugs, curtains and skylight screens not made of canvas, table covers, table linen, china, glass and plated tablewear, cooking utensils, crew's mess utensils, boat outfits, mail pouch, coaling baskets, and all supplies (Title "Y") except electrical supplies and such as pertain to articles installed by the contractor, but excluding cordage, canvas, etc.

(b) The ship is to be completely wired by the contractor for electric light and power, with all necessary fixtures, fittings, dynamos, engines, bed plates, wiring, instruments, tools, spare parts for engines, dynamos, etc., electrical stores, and searchlights.

116. The total weight of the equipment outfit complete, including the generating sets, must not exceed 300.26 tons.

117. The following articles will be furnished by the Government:

Boats and their spars.

Blocks.

Portable furniture.

Mess tables and benches.

Cooperage.

Anchorage, chains, cables, and all other equipment articles mentioned in paragraph 115 (a) of this circular.

Ordnance and ordnance outfits of all kinds.

Stores of all kinds.

118. General dimensions, etc., of the Department design:

Length on load-water line, feet	435
Breadth, extreme, at load-water line, feet	76
Trial displacement, tons, about	14,650
Mean draught at trial displacement, feet, about	24
Greatest draught, full load, feet, about	26
Total coal bunker capacity, tons	1,900
Coal carried on trial, tons	900
Feed water carried on trial, tons	66
Speed, knots not less than	19

119. Summary of weights to be carried, Department's design:

	Tons.
Guns, mounts, shields, etc.	805.00
Ammunition, ordnance stores and outfit (two-third full supply)	408.00
Steam engineering, complete, with water in boilers, condensers, piping, etc., and stores, etc., not to exceed	1,730.00
Reserve fresh water for steaming purposes (full supply)	100.00
Total coal capacity	1,900.00
Boats and outfits	42.00
Masts and spars	24.29
Equipment, complete, including anchors, chains, electric plant, etc.	300.26
Miscellaneous stores and water (two-third supply)	122.42
Officers, crew and effects	89.65
Total protection, including teak backing, armor bolts, cellulose, and splinter bulkheads	3,771.00

120. The work of constructing these vessels shall be done in accordance with this circular and the plans and specifications to be provided or adopted by the Department; but in any case where there may be a discrepancy between said specifications and this circular, the specifications shall prevail.

Kearsarge.—The final trial of this battleship took place April 3–5, 1900, and was, in many particulars, very gratifying and satisfactory. Extracts from the report of the trial board, as published in the "Army and Navy Journal," are appended:

1. In obedience to the Department's order of March 21, 1900, the board has the honor to submit the following report on the final trial of the U. S. S. *Kearsarge*. * * *

Ship's draught in getting underway was: Forward, 23 feet 2 inches; aft, 24 feet 0 inches; mean, 23 feet 7 inches.

4. The weather was clear and pleasant in getting underway, and during the entire day there was a haze around the horizon, with light to gentle breezes from northward to westward, shifting to southward and westward during the afternoon. The sea was smooth. During the second day (April 4, 1900) the weather was overcast and cloudy, with light air generally from southward to westward until 1·30 P. M., when the wind shifted to northward in fresh squall with rain. Fresh to strong breezes from northward the remainder of the day with a moderate sea. April 5 began cloudy, but cleared towards 2 A. M. The wind

remained strong, blowing in heavy squalls about daylight. Sea moderate.

5. At 1 P. M. on April 3 the two-hour full-power speed trial with natural draft was begun, with fires under all boilers, and with one representative of the contractors in the engine room. The coal used was Pocahontas of fair quality, containing a large proportion of slack. The speed made was 14.99 knots per hour. The machinery, both main and auxiliary, worked very satisfactorily, with the exception of a heavy steam leak in starboard H.P. piston-rod stuffing box. The boilers steamed freely.

At 3 P. M. the full-speed trial ended. The following is the data for the two-hours full power trial: Average revolutions, starboard, 98.7; port, 96.9. Vacuum, starboard, 26 inches; port, 25.5 inches. Average steam pressure, 154 pounds. Coal per hour, 18,480 pounds. Indicated horsepower, 8,483.35. Main and auxiliary engines, pounds of coal per horsepower, 2.18. Speed by patent log 1 to 2 P. M., 14.18 knots. Speed by patent log to 3 P. M., 15.80 knots. Average speed, 14.99 knots.

During the afternoon of April 3 the four guns in the forward turret were fired simultaneously at 4,000 yards range and three of the projectiles apparently fell in the same spot; one fell about 300 yards beyond, but in line. The four guns from the after turret were also fired simultaneously, and in this case three projectiles apparently fell in the same spot; one 8-inch projectile fell to the right. During this test the blast from the 8-inch guns in the superposed turret did not inconvenience the people in the 13-inch turret. Careful observation in the forward turret structure showed the following horizontal deflection of upper portion of turret support, the measurements being taken at level of turret roller path and immediately in rear of guns, the fixed point of reference being on the armored barbette.

Deflection due to firing simultaneously two 13-inch guns and two 8 inch guns, $\frac{7}{16}$ inch; for one 13-inch gun, $\frac{5}{32}$ inch. Rotation of turrets under this condition, measurement being taken at roller path, 9 inches. Deflection for one 8-inch gun, $\frac{1}{16}$ inch. Rotation of turrets under this condition, measurement being taken at roller path, 9 inches. Deflection for one 8-inch gun,

$\frac{1}{16}$. Rotation of turret under this condition very slight, but approximately $\frac{1}{2}$ inch.

The recoil of the forward turret, due to the firing all four guns simultaneously, caused the socket of turret securing the bolt to slightly foul the inner face of barbette, the armor at this point being uneven and projecting inward more than a quarter of an inch further than at any other point. As the turret support regained, completely and immediately, its normal form, the damage resulting from this contact was confined to the supporting socket, and such casualties in the future can be eliminated by removing an inch layer from the outer surface of the socket. It may be remarked here that the wedges and turnbuckles on main deck are amply sufficient to secure the turret, and that the principal use of the bolt in turret structure is for centering turret temporarily in drilling.

After these tests all the guns of the main and secondary batteries were fired at extreme elevation and at level. Owing to the range of the 13-inch guns and the haziness of the weather, these guns were fired at 10,000 yards instead of extreme elevation. No injury was done to the structure of the ship or to the gun mounts by these tests except as follows:

At first discharge of the left 13-inch gun of the after turret, the rear right cap square bolt was broken. On a previous occasion the front right cap square bolt of this gun had broken, and the Newport News Company made two new bolts. The broken bolt was replaced and no further accident occurred. On firing the automatic one-pounders in the military tops the vibration and jump to the mount was so great as to make any further firing of these guns dangerous.

7. The Ardois signal system and the search lights tried after dark of the 3d. The signal system worked well. The search lights were tested, but failed to give a satisfactory light. As these lights were put down in the equipment list of unfinished work as not finally adjusted, it is assumed this will be done later.

8. On the evening of April 3, 1900, the starboard anchor was let go in 28 fathoms of water and chain veered to 45 fathoms. The anchor was then hove up at speed to test windlass. The

chain came in at the rate of nine fathoms per minute, the anchor a-weigh.

9. April 4 was devoted to the detail inspection and tests of the structure and mechanisms of the vessel. The turrets were turned and guns elevated and depressed. The lifeboat was lowered. The detaching hook of forward fall failed to work and the block was unhooked by hand.

At 2·45 P. M., April 5, 1900, the *Kearsarge* anchored in Hampton Roads, having completed the 48 hours at sea. After anchoring in Hampton Roads, the cranes and winches were tested and found satisfactory.

11. The Board having been officially notified that the installment of the electric plant had not been fully completed, it was decided not to subject the turret ammunition hoists to the extreme tests required.

12. The Board considers the turrets unsufficiently lighted, and recommends the installation of an additional receptacle for a portable, and also the placing of a small incandescent light in each hood of the turrets, in order that the sights may be properly read. There is nothing in the 13-inch turrets now to indicate when the guns are in the loading position, or when they are at extreme elevations or depression. An index and scale is recommended to be fitted for each gun, by means of which the operator can readily ascertain the exact position in elevation of the gun, and a gong should be fitted in each turret near the voice tube in order that the signals may be received from the handling rooms. A ready means of communication from turrets to dynamo room should be installed. The telephones through central station cannot be relied upon on account of noise of turret mechanism.

It is also recommended that the valve stems controlling drainage of main hold compartments at forward and after end of ship be extended to deck above, so as to be operated from handling rooms of 13-inch turret. This change is considered necessary only for the large compartments, in order that the safety of the ship may not be jeopardized by their being flooded at a time when the present manifold box is inaccessible by reason of broken

steam pipe in engine room, an accident of this kind having occurred on the U. S. S. *Concord.*

14. On examining the ship and its fitting and the plans and detailed information prepared for the use of the ship's force by the superintending naval constructor, the Board was impressed with the completeness of the information already furnished and in course of preparation, and the many evidences of unusual care in inspection during the progress of the work at the shipyard; these results could only have been accomplished by a complete and thorough trained force, and the Board is of the opinion that all expenditures for the employment of skilled assistants in such work is many times repaid to the Government, both in greater excellence of original workmanship and more complete information for those subsequently charged with the care and management of the vessel.

15. While the Board was much impressed with the cleanliness and generally good condition of the ship, which reflected great credit upon the officers and men, and was the result of unremitting work on their part, it desires to specially bring to the Department's attention the fact that the benefits intended to be obtained by the Government as a result of the final trial of the vessels were not forthcoming, the vessel being, to all intents and purposes, fresh from the hands of the builders; in fact, many items of work were still incomplete, and, instead of the Department having the benefit of five months' test of the vessel and all of its appurtenances and fittings and service conditions, with the consequent opportunity to develop defects and remedies for the same, the Trial Board was inspecting a new and untried vessel under conditions, hull, machinery, etc., which more properly pertained to a first acceptance trial than those which should be required for the final acceptance trial.

In order, therefore, that the Government may obtain all the advantages to be derived from several months' actual trial of a vessel under service conditions, the Board urgently recommends that the first official trial be not made until the vessel is practically completed in all particulars, so that the final acceptance trial may furnish the Department with all needed data as to defects,

needed improvements, etc., developed by actual experience and under service conditions during the interval between the first and final trials:

Finally, the Board reports:

(1.) That during the trial no weakness or defects in the hull, fittings or equipment, due to either defective workmanship or defective materials, or in the fittings, fixing, placing and securing of the armor, due to defective workmanship, did appear, except as noted in the report and in the list of unfinished work appended.

(2.) That the machinery and all parts thereof was found to be in good condition with respect to workmanship or materials, and that there was no failure, breakdown, nor deterioration, other than that due to fair wear and tear, of any part or parts of the machinery, engines, boilers or appurtenances, except as noted in the report.

Isla de Cuba and **Isla de Luzon.**—These two captured gunboats which, after being sunk and partially burned at the battle of Manila, May 1, 1898, have been thoroughly overhauled at the works of the Hong Kong & Whampoa Dock Co. at Kowloon, Hong Kong, China, and both have made their four-hour official runs off Hong Kong. Forced draft was not used owing to the condition of the boilers and unreliability of the air pressure gages, but the fire-room blowers were run, discharging into the open fire-rooms. The coal used the on trial of the *Luzon* appeared to be of an inferior grade, making the consumption per square foot of grate much smaller than was the case with the coal used on the trial of the *Cuba*, so that the comparison of the performances is not at all a fair one, neither are the results in either case satisfactory nor indicative of full power attainable, owing to not using forced draft. The ships' displacements were also widely different. It must also be said that the revolutions of the engines of the *Isla de Cuba* were attained by interval counting, the regular counters being out of order.

The following data applies to both ships, and is given as a preliminary to the trial data:

GENERAL DATA.—*ISLA DE CUBA* AND *ISLA DE LUZON.*

Where built	Sir W. G. Armstrong, England.
When launched	1887
Length between perpendiculars, feet	185
over all, feet	197
Beam, extreme and L.W.L., feet	30
Area immersed midship section, square feet	300
Mean draught (load line), feet and inches	11-6
Displacement (load line), tons	1,030

Main engines, twin-screw, triple-expansion, horizontal, direct-acting.

Number cylinders each	3
Diameter cylinders, inches, H.P.	18½
I.P.	29
L.P.	43
Stroke, inches	24

Main Condenser.

Each cooling surface, square feet	1,665
Total cooling surface, square feet	3,330
Cooling surface to heating surface	1 to 1.654

Main Air Pumps (attached to main engine).

Diameter, inches	12
Stroke, inches	24

Main Circulating Pumps.

Type	centrifugal, with vertical, single steam cylinder.
Diameter steam cylinder, inches	5
Stroke, inches	3
Diameter of runner, inches	24
water suction, inches	7
discharge	7

Main Boilers.

Number	2
Type	Gunboat.
Diameter, external, feet and inches	10-6
Length, external, feet	19
Furnaces in each, (plain cylindrical in two sections with Adamson rings)	4
Total number of furnaces	8
Diameter of furnaces, internal, inches	33
Length of grate, feet and inches	6-9
Working pressure, present allowed, pounds, gage	140

	One Boiler.	*Total.*
Grate surface, square feet	74.25	148.5
Plate-heating surface, square feet	381.6	.....
Tube-heating surface, square feet	2,372.7	.....
Total heating surface square feet	2,754.3	5,508.6
G.S. to H.S.	.....	1 to 37
Number of tubes, ordinary	356	712
stay	61	122
Length of tubes between tube sheets, feet and inches		7–10⅞
Diameter of tubes, outside, inches		2¾
Smoke pipe, area, square feet		29.06
One smoke pipe, height above grate, feet		54.25

Screw Propellers.

Number	2
of blades each	3
Material	Bronze.
Adjustable blades.....No.	Blades keyed in hub.
Diameter, feet and inches	9–6
Pitch, mean, feet and inches	10–6
Disc area, square feet, each	66.75
Helicoidal area, square feet, each	21
Immersion of upper edge, load draught, inches	27

Trial data.	*Isla de Luzon.*	*Isla de Cuba.*
Date of trial	Jan. 25, 1900.	March 6, 1900.
Duration of trial, hours	4	4
Mean draught, feet and inches	11–4¾	10–6
Displacement, tons	1,020	950
Speed, knots per hour	11.225	12.89
Revolutions main engines, per minute, starboard	113.7	134.7
port	114.5	130
Vacuum, inches, starboard	20.75	21.2
port	24	21.8
I.H.P. main engines, starboard	232.8	353.6
port	266	465.7
total	498.8	819.3
I.H.P. all machinery	534.8	851.04
Condenser cooling surface, total, square feet	3,330	3,330
Grate surface in use, square feet	148.5	148.5
Heating surface in use, square feet	5,508.6	5,508.6
Coal used, pounds per hour	1,840	3,086
per square foot grate surface, pounds	12.4	20.7
per I.H.P. (total) per hour, pounds	3.44	3.62
Heating surface per I.H.P. per hour, square feet	10.3	6.47
Cooling surface per I.H.P. per hour, square feet	6.22	3.92
I.H.P. per square foot of grate surface	3.6	5.7

Holland.—This submarine torpedo boat has after a number of very successful trials, been purchased by the Government and is to be utilized at Newport torpedo station for instructing officers and men in its handling.

The contract was made on April 11, 1900, the price paid being $150,000. By the terms of the contract the Government has the privilege of ordering additional submarine boats of this type at a price not exceeding $170,000 each and of different dimensions from the *Holland*, according to plans submitted by the Holland Torpedo Boat Co. for *New Holland*.

New Revenue Cutter.—Bids for the construction of a new revenue cutter for the Great Lakes were opened at Washington recently as follows: American Shipbuilding Co., Cleveland, delivery on or before May 1, 1901, $150,000; Chamblin & Scott, Richmond, Va, delivery in twelve months, $163,500; Townsend & Downey, New York, delivery in twelve months, $151,000; William R. Trigg Co., Richmond, Va., delivery on or before May 28, 1901, $157,000. This was the first appearance of the second-named bidders in competition.

The new revenue cutter is to be a steel, single-screw vessel of about 620 tons displacement. Her general dimensions are: Length over all, 178 feet; beam, molded, 30 feet; depth, to base line, 15 feet. Captain Russell Glover, R. C. S., superintendent of construction, designed the hull. This will be of mild open-hearth steel of a tensile strength of 55,000 pounds per square inch, with an elongation of 25 per cent. in 8 inches. Physical tests to be applied require that the material must withstand bending traversely back on itself without sign of fracture. This grade of material has been found satisfactory in every way for use in the construction of the hulls of revenue cutters. The frames are made of 5-inch by 3½-inch by 3½-inch Z-bars, spaced 2 feet centers and secured to a vertical keel plate of 15 pounds steel, about 21 inches deep. The floor plates generally will be 18 inches deep next to the vertical keel, while in the engine and boiler space they will be extended up so as to form a part of the foundations for the engine and boiler. The outside plating up to the line of the main deck will be of steel, 15 pounds per square

foot. The stem will be slightly ram shaped. The stem and stern post will be forgings of wrought iron.

The hull is divided into compartments by four watertight bulkheads. The quarters for the commissioned officers are located on the berth deck aft of the engine compartment, and the quarters of the petty officers and crew are forward of the boiler compartment. The accommodations for the crew, as well as for the officers, are comfortable, and every care has been exercised to make the conditions of life on board ship as pleasant as possible. To this end the sanitary system of the vessel has been made very complete, shower baths and running water being provided for the crew, and the water closets being continuously flushed.

The pilot house and chart room are to be located on the forecastle deck forward of the smoke pipe, and a bridge will extend across the top of the pilot house to the ship's side.

Two bower anchors will be provided, each weighing about 1,800 pounds. They will be secured to 1¼-inch stud-link chain, provided with shackles every 15 fathoms. The anchors are to be of the stockless type.

Provision will be made for carrying a certain amount of sail principally to be used for steadying purposes in a seaway. For this purpose the vessel will have two steel masts, each about 80 feet high above the main deck. The propelling machinery of the vessel, which it is expected will give her a speed of 14 knots at full power, was designed by Captain Jno. W. Collins, R. C. S., engineer-in-chief, who also has charge of the electric-light, sanitary and steam-heating systems, and the inspection of all material.

A triple-expansion engine, designed to develop 1,200 I.H.P., at 195 revolutions, with 175 pounds initial pressure, will be fitted. This engine has cylinders 17 inches, 27 inches and 43 inches diameter and 24 inches stroke. A piston valve will be fitted to the high-pressure cylinder, and double-ported slide valves for the other cylinders. All boxes will be fitted with Katzenstein's metallic packing.

Steam will be generated in two Scotch boilers, each with three

corrugated furnaces. The total grate surface of each boiler is about 55 square feet, and the total heating surface about 1,650 square feet. The shells will be 12 feet diameter and $1\frac{1}{8}$ inch thick, each made up of two plates. The overall length of the boilers is 10 feet 3 inches. The horsepower per square foot of grate surface is expected to be about 11.

The condenser will form a part of the framing of the engine, and will contain about 1,350 square feet of cooling surface, made up of 793 Muntz metal tubes $\frac{5}{8}$ inch outside diameter, No. 16 B.W.G.

The air pump will be an independent, single-acting, twin air pump secured directly to the condenser. It will have two steam cylinders, each $7\frac{1}{2}$ inches diameter, two air cylinders, each $16\frac{1}{2}$ inches diameter, stroke being 10 inches.

The circulating pump will be of the centrifugal type, having 8-inch suction and discharge, and directly connected to a 7-inch by 7-inch vertical engine. The pump will be connected to the sea by two injection valves, and to the bilge by a bilge injection and a Macomb strainer.

A feed-water heater is to be provided, having about 120 square feet of surface. This heater will receive the exhaust of the auxiliaries on the outside of the tubes, the feed water passing through from the feed pump to the boiler. The heater will be provided with a combined air and circulating pump, so arranged that when the vessel is not under way the heater can be used as an auxiliary condenser for such auxiliaries as require to be kept continuously in operation.

The feed pump is to be of the duplex pattern, having steam cylinders $7\frac{1}{2}$ inches diameter, water cylinders $4\frac{1}{2}$ inches, and a stroke of 8 inches.

Besides these auxiliaries, the vessel will be fitted with steam-steering gear, steam windlass, fire and flushing pumps, etc., so as to make her thoroughly efficient in every way.

The auxiliaries are to be connected so as to exhaust into the main condenser, the heater or the receiver at will. By the latter method an economy approaching that of the main engine can be

obtained for the auxiliaries, and still all the advantages of separate auxiliaries retained. The propeller will be a solid cast-steel wheel, 8 feet diameter.—"Marine Engineering."

ENGLAND.

New Ships.—The new programme of shipbuilding provides that there shall be laid down during the coming year two battleships, six first-class armored cruisers, one second-class cruiser, two sloops, two light-draught gunboats and two torpedo boats. Of these, the two battleships, two of the armored cruisers, the second-class cruiser and the two sloops will be built in the dockyards; so that, putting aside the four boats, only four cruisers are be given out to contract. Considering the failure of contractors to meet the programme, it was to be expected, and, indeed, inevitable, that the major part of the new construction should be carried out by Government establishments. On the whole programme, however, the value of work to be done under Vote 8 is almost equally divided between the dockyards and the contractors, the former being allotted construction which is to cost 6,596,000 pounds, while there is to be spent on contract work 6,329,000 pounds. The full list of ships that will be in hand is one of considerable magnitude, consisting of no less than 77 vessels of various sizes, of which 17 are battleships and 20 armored cruisers. The other 40 are made up by one first-class protected cruiser, two second-class protected cruisers, one third-class cruiser, eight sloops, two light-draught gunboats, four torpedo boats, 21 torpedo-boat destroyers, and one royal yacht. The whole forms a list that would be a respectable navy for most foreign powers; but it is not a ship too many.—"Engineering."

Kent.—The armored cruiser *Kent*, which has just been laid down at Portsmouth, is a sister ship to the *Essex*, recently laid down at Pembroke, and the *Monmouth* and *Bedford*, which have been put out to contract, may be described as a modified *Leviathan*, one of the "mighty cruisers" described by Mr. Goschen in introducing the Navy Estimates a year ago. She is to be 440 feet in length and 66 feet in breadth, with a displacement of 9,800 tons, and will be armed with fourteen 6-inch quick-firing guns, ten of

which will be in casemate, and two forward and two aft on twin mountings. She will also have six 12-pounder quick-firing guns amidships on the main deck, and two forward and two aft on the upper deck. Thus on a 1,200-ton less displacement she is provided with two fewer 6-inch and four fewer 12-pounder guns than the *Diadem* class; but she has many compensating advantages, as not only will she be armored, but she will have a speed of 23 knots, against 20½ knots in the *Diadem*. Her 4½-inch armor will begin 106 feet from the stern, and, forward of the vital parts, will taper off to 2 inches at the bows. The after end will have a 2-inch armored deck, and there will be two protective decks, the thickness on the main deck being 1½ inches and the lower deck ¾ inches. The thickness of the casemate armor will be 4 inches. Protection to the fore and after guns will be furnished by a 4-inch barbette surmounted by a shield of similar thickness, revolving with the guns, which are to be carried on Vickers mountings.

The *Kent* will carry about 1,500 tons of coal, or 500 tons more than the normal bunker capacity of the *Diadem*, and, in order to provide for the extra weight of armor and coal, the ship has been designed throughout to produce a combination of lightness and strength. There will be no boat deck, for instance, the boats being carried on skid beams on the upper deck, and the protective decks will be thinner than in the *Diadem* class, while there will be one casemate less on each side. The complement will be about 600 officers and men. There will be 31 Belleville boilers in three boiler rooms, with six stokeholds in three compartments, the longitudinal space between the protective deck occupied by the engines and boilers being 194 feet, of which one-third will be taken up by the engines. She will draw 24 feet forward and 25 feet aft in ordinary seagoing trim, though with all the stores on board she will be somewhat lower in the water; and she will have a freeboard of 19 feet amidships. Ventilation will be afforded by means of electrical fans, as in the *Formidable* class, each compartment between the main bulkheads being separately ventilated by electrical-motor fans. The *Kent* will carry two masts, but no fighting tops. She is the longest

ship ever laid down at Portsmouth, and the building slip has been lengthened 100 feet in order to take the keel.

Glory.—Her Majesty's first-class battleship *Glory*, of 12,950 tons displacement and 13,500 indicated horsepower, built and engined by Messrs. Laird Brothers, of Birkenhead, has just completed her official trials in the English Channel. On these trials the Admiralty were represented by Messrs. W. J. Berry (Assistant Constructor) and J. H. Ellis (Engineer Inspector) and the Dockyard by Messrs. Godbear and Roberts, the vessel being in charge of Captain Parr, of the Steam Reserve. The contractors were represented by Mr. Roy M. Laird and Mr. R. Ratsey Bevis, Jr. A thirty hours' trial at one-fifth of the contract power, and another thirty hours' trial at about four-fifths of the contract power, having previously been satisfactorily completed, the eight hours' full power trial took place on February 23. It was decided to make this trial over the new 25-fathom course recently plotted out by the Portsmouth Reserve, and the vessel proceeded to Portland on the 22d, having previously completed her anchor trials at Spithead, and anchored there for the night. Proceeding next day, the eight hours' trial was commenced at 8 A. M. Four runs were made on the course off the Start in the teeth of a moderate westerly gale, and under the somewhat adverse circumstances a speed of 18.124 knots was obtained. The vessel was then headed up channel, and throughout the entire trial the engines and boilers worked satisfactorily. Messrs. Laird are to be congratulated on the success of the working of the boilers, which are the first installation of Belleville boilers that they have officially tried. The results of the trial were as follows: Mean steam in boilers, 275 pounds; vacuum, 27 inches starboard, 26 inches port; revolutions, 108.5 starboard, 106.7 port; indicated horsepower, 7,021 starboard, 6,724 port; total mean indicated horsepower, collectively, 13,745; coal consumption, 1.58 pounds per indicated horsepower per hour.

It may be interesting to state the results of the thirty hours' trial at 10,250 indicated horsepower, which had taken place a few days previously: Steam, 250 pounds; vacuum, 27.4 inches starboard, 26.4 inches port; revolutions, 99.4 starboard, 94.2

port; indicated horsepower, 5,253 starboard, 5,334 port; total, collectively, 10,587; coal, 1.7 pounds per indicated horsepower per hour; mean speed, 16.78 knots. After finishing the eight hours' full-power trial, the usual stopping, starting and reversing trials, etc., were completed. The principal dimensions of the *Glory* are: Length, 390 feet; breadth, extreme, 74 feet; displacement at load draught, 12,950 tons. Her propelling machinery consists of two sets of triple-expansion engines, each having three vertical cylinders of 30 inches, 49 inches and 80 inches in diameter respectively, with a piston stroke of 51 inches. They each drive a four-bladed gun-metal screw propeller. The engines are designed to develop 13,500 indicated horsepower at full power. Steam is supplied by 20 water-tube boilers of the Belleville type, consisting of 15 generators of nine elements and five of eight elements, with an economizer to each boiler. The total heating surface of generators and economizers is 33,700 square feet, and the grate surface 1,055 square feet. The *Glory* is the first battleship of the *Canopus* type built by private contract to be delivered, and there is no doubt that the completion would have been anticipated but for the engineers' strike, and the delay in obtaining deliveries of material of all kinds, particularly the armor. The vessel will now be rapidly brought forward for commission, and her gun trials will take place shortly. —"Engineering."

Seagull.—The *Seagull*, torpedo gunboat, Commander H. Grant-Dalton, which is the only ship in the service fitted with the Niclausse water-tube boiler, has concluded a series of nine trials, each of approximately 1,000 miles. Four of the runs fell short of the required distance, owing to the bad weather. At the four early trials the indicated horsepower ranged from 1,354 to 1,371, and the speed varied from 13 to 13.6 knots. The next trial gave a speed of 14.48 knots, with 1,611 indicated horsepower, but on the following trial with an additional 20 indicated horsepower, the speed went up to 14.6. The next trial was carried out in very bad weather, and had to be abandoned when only 855 miles had been run; but with 1,798 indicated horsepower the *Seagull* averaged 15.2 knots. The next run was in such fine

weather that the ship was able to complete the thousand miles, and then with two additional indicated horsepower her speed improved by two points. At the final trial, with 1,947 indicated horsepower, the speed was 16.07 knots. Throughout the trials only four of the six boilers were in use, and as the engines are capable of 3,000 indicated horsepower, the actual power at the time of the last trial was, for the number of boilers in use, $\frac{19}{20}$ of the maximum. The maximum coal consumption for the entire series of trials works out at 1.9 pounds per unit of power per hour for the main engines, and two pounds for all purposes. The Niclausse type of boiler has been shown to possess certain advantages, the chief of which is that any tube can be quickly removed and another substituted, while all the tubes, which are $3\frac{1}{4}$ inches in diameter, are straight. When the trials were begun, the lanterne, or end of the tube, was screwed on, but now the tube and lanterne form one solid-drawn piece, which materially facilitates the substitution of tubes.

Trial of the Highflyer and Minerva.—The results of two trials of the *Highflyer* and *Minerva* (second-class cruisers of 5,750 tons) are as follows:

March 3, 1900. *Minerva* and *Highflyer* completed satisfactorily C. 2 steam trials (*i. e.*, of 60 hours' duration at 17 knots). Coal, all purposes, *Minerva*, 358 tons; *Highflyer*, 375 tons. Coal per indicated horsepower, engines only, *Minerva*, 1.95; *Highflyer*, 2.07. Coal per indicated horsepower, all purposes, *Minerva*, 2.22; *Highflyer*, 2.37.

March 7, 1900. *Minerva* and *Highflyer* completed satisfactorily D. 1 steam trials (*i. e.*, of 30 hours' duration at highest speed that can be maintained. Coal, all purposes, *Minerva*, 225 tons; *Highflyer*, 267 tons. Coal per indicated horsepower, engines only, *Minerva*, 1.97; *Highflyer*, 2.1. Coal per indicated horsepower, all purposes, *Minerva*, 2.22; *Highflyer*, 2.52. *Highflyer* averaged 400 horsepower and $\frac{1}{2}$ knot greater speed than *Minerva*.

It will be recalled that these trials were for comparative efficiency of cylindrical and water-tube boilers, the *Highflyer* having water-tube boilers and the *Minerva* Scotch boilers.

Shearwater and Vestal.—The new sloops *Shearwater* and *Vestal* have been launched; they are sister vessels to the *Condor* and *Rosario*, which were built in the same dock in 1898. Designed by Sir W. H. White, K. C. B., Director of Naval Construction, they have each a length of 180 feet, a beam of 32 feet 10 inches, a draught forward of 10 feet, aft, 13 feet, and a displacement of 980 tons. Their steel plating is a quarter inch thick, sheathed with wood four inches thick, to a height of 2 feet above the load draught. The copper sheathing extends to a foot above the water line. The holds of the ships are minutely divided into watertight compartments, while above the boiler and engine room runs a steel watertight deck, which forms a division between the upper and lower coal bunkers. When completed the sloops will carry square yards on the foremast and mainmast, and will be the exact counterparts in appearance of the *Rosario*. They will be armed with six 4-inch Q. F. guns, four 3-pounder Hotchkiss guns and two .45-inch Q. F. Maxims.

Condor.—The new sloop *Condor* has satisfactorily completed the whole of her steam trials, and, except for the little hitch when her piston rod became overheated and she had to be towed back into harbor, she has gone through the ordeal most successfully, and in one or two respects with even better results that her dockyard-engined sister, the *Rosario*. The eight hours' trial of the *Condor* gave an average I.H.P. of 1,462.5, being 62.5 horsepower more than was required by the contract. This was 35.5 horsepower less than the mean of the *Rosario's* machinery; but the mean steam pressure in the boilers of the *Condor* was 231.7 pounds, compared with 235.4 pounds in the *Rosario*. The engines of the *Condor* worked 197.1 revolutions per minute, while the *Rosario's* average was 235.3. The difference in speed was infinitesimal, the *Condor* averaging 13.68 knots per hour, and the *Rosario* 13.6 knots. The *Condor* had an advantage over her sister sloop in matter of coal consumption, her average being only 1.55 pounds per I.H.P. per hour. The *Condor* is to be finished by the end of the financial year, but, as in the case of the *Rosario*, overtime will have to be extensively resorted to in order to insure this.

Rosario.—The new sloop *Rosario* has also completed her trials. The results of her eight hours' full-power natural-draft trial are appended: Pressure of steam in boilers, 235.4 pounds; at engines, 185.4 pounds; revolutions, 205.3 per minute; I.H.P., high, 400.5; intermediate, 524.5; low, 573; total, 1,498. The contract provided for 1,400 horsepower, so that there was a margin of 98 to spare. The speed of the *Rosario* was exactly that estimated by her designer, viz: 13.6 knots per hour. The machinery of the *Rosario* was made at Devonport.

Petrel.—The tenth 30-knot torpedo destroyer, the *Petrel*, built and engined by Palmer's, underwent her three hours' consumption trial from Portsmouth dockyard March 30. An average of six runs over the measured mile gave a speed of 30.322 knots, with 384.6 revolutions, and 240 pounds steam pressure, the indicated horsepower being 6,632, and vacuum 27 inches. For the three hours the speed aimed at was 30.1 knots, the intention being only to exceed the 30 knots guaranteed. The results were: Speed 30.097 knots, 380.6 revolutions, 239 pounds of steam, 6,438 indicated horsepower, and 27 inches of vacuum. The trial was run on a heavier displacement than any of Palmer's previous 30-knot boats, and this enhances materially the value of the present performance.

Ostrich.—The Fairfield Shipbuilding and Engineering Company, Limited, launched at Govan, the 30-knot torpedo-boat destroyer *Ostrich*. This destroyer is sister ship to the *Falcon*, launched by the same company about three months ago, and now being prepared for trials, and is the ninth of this class of vessel built by them for the British Admiralty.

H. M. Torpedo-Boat Destroyer Viper.—On Friday, May 4, the official trial of the torpedo-boat destroyer *Viper* was made on the North Sea, off the mouth of the Tyne. As our readers are aware, this vessel possesses unusual interest from the fact that her main engines consist of Parsons' steam turbines.

The *Viper*, in general arrangement, is similar to the later class of torpedo-boat destroyers built for Her Majesty's Navy. She is 210 feet long, 21 feet wide, and 12 feet 9 inches deep. The displacement on trial was 370 tons, with all coal on board. The

contract load was 40 tons, but, as a matter of fact, the vessel had on board 60 tons. There are four shafts, each having two propellers, or eight in all. The propelling machinery consists of two sets of compound steam turbines for driving the vessel ahead.

As the steam turbine cannot be reversed, special turbines are provided for going astern; the speed in that direction being stated to be 15½ knots.

The vessel herself has been built by Messrs. Hawthorn, Leslie and Co. at their Hebburn yard, the same firm having constructed the four Yarrow boilers by which steam is generated.

The contractors have been exceptionally unfortunate—even for the northeast coast—with this vessel, as she has been waiting her trials for a long time. Time after time the date has been fixed, only to find when the day arrived that a gale was blowing outside, or, at any rate, an onshore wind strong enough to raise a sea that would prevent the vessel being run at anything like her exceedingly high speed, so that the postponements have been numerous.

The *Viper* ran as far as the measured mile with the wind and sea quarterly. Full steam had not been reached when she entered on the mile, and even on the second run the speed did not quite reach 30 knots. Ten runs were made in all, and taking the best six consecutive runs—of course with and against tide—a speed of 34.28 knots was reached. The best two runs gave a mean speed of 34.75 knots, while the mean speed on a three-hours' run was 33.96 knots.

The mean revolutions were 1,050 per minute, and the mean steam pressure was 165 pounds to the square inch, the maximum being 175 pounds. In regard to this point it should be stated that a great deal of steam was lost at the relief valves, which had not been sufficiently screwed down before starting, and began to blow heavily soon after going on the mile. The total heating surface in the four Yarrow boilers is 15,000 square feet, and the grate surface is 275¾ square feet. The air pressure for draft averaged 3 inches on the water gage.

The indicated horsepower, estimated by the steam pressure

and revolutions, is taken at 11,000, which shows an excellent performance, even allowing for the efficiency of the engine. The boat had been in the water for some time, and her bottom was said to be by no means clean. The officials of the Parsons' Company expressed themselves confident of being able to get 36 knots with a clean-bottom boat on a fine day, provision being made, of course, for using all the steam and not allowing any to pass away at the relief valves.

The weight of the steam turbine as compared to ordinary engines is as 35 to 53; whilst in regard to space occupied it may be said that the engine room of the *Viper* is the same size as that of the other vessels of her class, but, of course, there is the additional power developed by the machinery of this boat to take into account. The size of the engine room is, however, hardly a fair basis of comparison, as the space is less occupied than usual. The turbine naturally lends itself particularly well to neat stowage. The diameter of the high-pressure cylinder is 35 inches, and that of the low-pressure cylinder 50 inches.—"Engineering."

Bat.—A serious mishap, fortunately unaccompanied with loss of life, took place on board the torpedo-boat destroyer *Bat* on the 21st of February. She was engaged on instructional class duty. The instructional classes have a course of four weeks; the first three weeks consisting of runs at gradually increasing speeds, culminating in one hour's run at top speed. In the case of the *Bat*, this full-power run was in course of being carried out. Everything was apparently going well, the hour was nearly up, the engines were making 360 revolutions and the boat was traveling at 28 knots. A loud noise was then heard, followed by an escape of steam. Steam was promptly cut off and the eight occupants of the engine-room at the time succeeded in getting on deck without injury. The starboard engine was found to be badly damaged, and the *Bat* returned to Devonport with the port engine only. On further examination it is stated that the prime cause can be traced to the cap of the low-pressure crosshead brasses, which fractured across one of the bolt holes. This fracture freed the connecting rod, which knocked against, and badly bent, the crosshead guide, and made a bulge in the

condenser. The piston being also free, was forced up and broke the cylinder cover, at the same time badly cracking the cylinder.

FRANCE.

Montcalm.—The new French cruiser *Montcalm*, which was successfully launched on March 27 by the Forges et Chantiers de la Méditerranée at La Seyne, is a reduction of the *Jeanne d'Arc*. Her length is 461½ feet; beam, 64 feet; and displacement, 9,516 tons. She has a 6-inch armor belt of Harveyed steel and a protected deck. Her armament consists of two 7.8-inch quick-firing guns in turrets, eight 6.6-inch quick-firing guns in casemates, four 4-inch with semicircular shields on the spar deck, six 2-inch and six 1½-inch distributed aloft, and two submerged torpedo tubes. She has three propellers and three vertical triple-expansion engines, supplied with steam by Normand-Sigaudy multitubular boilers, and developing 19,600 horsepower, giving her a speed of 21 knots. Her radius of action at 10 knots exceeds 10,000 miles. The *Dupleix*, French cruiser, which was launched at Rochefort on March 28, is intended for distant stations. She is smaller, but of the same design as the *Montcalm*. Her length is 429 feet; beam, 59 feet; and displacement, 7,800 tons. The thickness of her armor belt is only 4 inches; it extends 4 feet below the water line, and 10 feet above. The steel deck is 2 inches in thickness. She carries ten 6.6-inch quick-firing guns, model 1893–'6, two in turrets and eight in casemates, while on the bridge and spar deck there are ten 2-inch and six 1½-inch quick-firing guns. Her two torpedo tubes are above water. The three engines are triple-expansion; they are supplied by 24 Belleville boilers, and develop 17,100 horsepower, giving a speed of 21 knots. Her radius of action at 10 knots is 8,400 miles. The *Yatagan*, French torpedo-boat destroyer, was launched at Nantes on March 20. She is of the same type as the *Framée*, now undergoing her trials at Lorient. Her length is 185 feet; beam, 19 feet 7 inches; displacement, 303 tons. She carries one 2.6-inch and six 1.88-inch quick-firing guns and two torpedo tubes. Her engines are 4,800 horsepower, driving two

propellers, giving her a speed of from 26 to 27 knots. Her radius of action is 3,000 miles.—"Le Yacht."

Submarine Boat.—The submarine boat *Morse*, after what are considered a series of very successful trials at Cherbourg, has been accepted by the naval authorities; she will be sent for a cruise soon to visit different places along the coast. She is supposed to have a radius of action of 100 miles. On her last full-speed trial she made 12.3 knots, a notable improvement on the *Gustave Zédé*, which has never attained a higher speed than 8 knots. The new submarine boat *Narval* has been trying at Cherbourg her torpedo discharges, which are on the Drzewiechi system, with, on the whole, satisfactory results.

Vessels Fitted with Normand-Sigaudy Boilers.—The new torpedo-boat destroyer *La Hire* has been carrying out some supplementary trials; she is fitted with the Normand-Sigaudy water-tube boilers, it being claimed for these boilers that they are very light, easily managed and quite safe, owing to the small amount of water which passes though the tubes, while the pressure can be very rapidly increased or diminished as required. These boilers have been fitted to the first-class cruisers *Chateaurenault* and *Jeanne d'Arc*, and the weight gained by their lightness has notably increased the amount of coal these ships are able to carry.

GERMANY.

Kaiser Barbarossa.—This new triple-screw battleship has been launched at the Schichau works at Danzig. Her principal characteristics are as follows: Length over all, 410 feet; length between perpendiculars, 377 feet; beam, 67 feet; displacement, 41,080 tons.

The side armor (Krupp's steel) extends four-fifths of the length from the bows and is 6 feet wide, varying in thickness from 6 inches to 12 inches. The protective deck extends the entire length of the vessel and is from 2½ to 3 inches thick.

The engines are of the four-cylinder triple-expansion type, of usual vertical direct-acting design, and the total horsepower is to be 13,000, with forced draft and a speed of 18 knots.

The normal coal supply is 650 tons, but this can be increased to 1,000 tons.

Her armament will be four 9.46-inch rapid-fire guns in pairs in two turrets, one forward and one aft, the turrets being 10 inches thick; eighteen 6-inch guns (twelve in casemates and six in turrets); twelve 3½-inch guns; twelve 1.45-inch guns; twelve machine guns; six torpedo tubes, five of these being submerged.

The complement will be 655 officers and crew.

Beowulf.—The coast-defense armored ship *Beowulf*, which last summer formed part of the maneuvering fleet, and since the dissolution of the latter has been the flagship of the reserve coast-defense battleship division in the North Sea, will be put out of commission at the Danzig Imperial Navy Yard about the middle of April for the purpose of undergoing extensive modifications. It is proposed to lengthen the ship by seven frames, as was recently done in the case of the battleship *Hagen* at the Kiel yards. In order to carry this in effect it will be necessary to remove all longitudinal joints between frames 30 and 31, and pull the two halves of the ship 8.4 meters apart on sliding ways. In addition, the *Beowulf* is to be equipped with new water-tube boilers, eight of which will be required. As far as possible, these boilers are also to be constructed at the Danzig imperial yard. Furthermore, all wooden decks are to be removed, with the exception of the upper deck abaft of the superstructure, and the forecastle. The wooden steps, and all woodwork that can be dispensed with in the ammunition rooms, holds, etc., will also be taken out and replaced by steel. It is proposed to complete the changes by April 1, 1902.

Hagen.—The work of lengthening the small coast-defense battleship *Hagen* amidships is now almost completed, and it is hoped that the ship will be ready for her trials by midsummer; 26 feet 10 inches has been added to her length and the additional room thus gained will be utilized to increase her coal supply, which, as the ship was originally designed, was only 250 tons; her displacement will be increased by something over 100 tons, and it is also hoped that the seagoing qualities of the ship will

also be improved. A sister ship, the *Beowulf*, is to be similarly treated and equipped with new water-tube boilers.

Water-Tube Boilers.—The Schulz system of water-tube boiler, which is a modification of the Thornycroft, is being more and more adopted in the German Navy. The battleship *Württemberg* has lately received these boilers, and they are to be supplied to the new cruiser *Nymphe;* the new battleships *Kaiser Wilhelm der Grosse* and *Kaiser Karl der Grosse* are supplied with one-half of their boilers the old cylindrical pattern, and the other half the Schulz, while the battleships *Kaiser Friedrich III* and *Kaiser Wilhelm II* have two-thirds of their boilers cylindrical and one-third of the Thornycroft pattern; the new armored cruiser *Fürst Bismarck* has two-thirds of her boilers cylindrical and one-third Schulz, while all the new ships will be fitted with boilers of this type.

JAPAN.

Asahi.—The battleship *Asahi*, built for the Imperial Japanese Navy by Messrs. John Brown & Co., Limited, at their Clydebank establishment, has just had a most successful series of trial tests. The battleship is the largest one in the Japanese Navy, and is of the following dimensions: Length between perpendiculars, 400 feet; length over all, 425 feet 6 inches; breadth, extreme, 75 feet 2½ inches; depth, molded, 43 feet 7½ inches; normal mean draught of water, 27 feet 3 inches; displacement, 15,200 tons; indicated horsepower, 15,000.

Four 12-inch guns of the most modern type are mounted in pairs in two barbettes, one forward and the other aft, on the middle line of the vessel, each pair commanding an uninterrupted arc of training of 240 degrees. The manipulation of the machinery and all the operations of loading and laying the guns are performed by hydraulic power; the loading can be performed with the guns in any position of training. The guns and gunners are well sheltered by means of heavy armored shields, which revolve with the turntables. The secondary armament consists of fourteen 6-inch quick-firing guns, each mounted in a separate casemate; twenty 12-pounder, eight 3-pounder and four 2½-

pounder quick-firing guns, with four submerged torpedo tubes in two compartments, one forward and one aft.

Great care has been bestowed upon the arrangement of the protective material. There is a main belt extending for a length of 250 feet amidships, the total depth of this belt being 8 feet 2 inches, and it is intended that when the ship is floating at the normal water line the lower edge of the armor will be 5 feet 6 inches below water, and the upper edge 2 feet 8 inches above water. The maximum thickness of this belt is 9 inches. The central citadel of armor is completed by transverses or bulkheads extending obliquely across the ship and enclosing the bases of the barbettes which protect the positions for the heavy guns. Forward and aft of this main belt the protection of the water-line region of the ship is completed by armor carried to the bow and stern. Above the main belt the sides from lower to main deck are covered with armor of a thickness of 6 inches; this belt extends for a length of 250 feet, and is completed by oblique transverses at the ends as described for the main belt. Armored doors are fitted in these transverses for affording convenient means of communication along the deck when in port. The protection of the vitals of the ship is rendered the more secure by a heavy protective deck extending all fore and aft, and sloping away from the underside of the main armor belt. The protection to the armament is arranged in a very thorough manner. For the 12-inch guns at the extremities circular barbettes rise from the protective deck to a height of 22 feet 4 inches above the normal water line; these barbettes are plated with armor of a maximum thickness of 14 inches. Each 6-inch quick-firing gun is enclosed in a casemate with an armored front 6 inches thick. The forward conning tower is composed of 14-inch armor, and the after tower of 3-inch armor. The whole of the armor plating was manufactured by Messrs. John Brown & Co., Limited, Sheffield, and is of the highest quality procurable, every plate having been treated by the improved Harveyed nickel-steel process, except those for the conning tower, where experience shows that the curvature of the plates is too great to enable this process to be successfully employed; these plates

are consequently made of ordinary Harveyed steel. As a protection against attacks from torpedoes, a broadside net defence is fitted for a length of about 300 feet amidships; the nets are supported by steel booms, and when not in use stow on a convenient shelf worked round the sides.

The ship is propelled by two sets of three-cylinder triple-expansion engines. Each of the two sets is designed to develop 7,500 indicated horsepower, giving a combined indicated power of 15,000. Steam is supplied by eighteen water-tube boilers of the Belleville economizer type, working at a pressure of 300 pounds, which is reduced at the engines to 250 pounds. The diameters of the high-pressure cylinders are 32½ inches, of the intermediate-pressure cylinders 52 inches, and of the low-pressure cylinders 85 inches, all having a stroke of 4 feet.

The vessel left Portsmouth Harbor on Tuesday, March 20, and after compasses had been adjusted, a few low-speed runs on the measured mile at Stokes Bay were made with the following results: Mean speed, 6.69 knots, indicated horsepower, 613; speed, 9.28 knots, indicated horsepower, 1,610; and for a speed of 13.06 knots the indicated horsepower was 4,355. On Wednesday, the 21st, the vessel ran her coal-consumption trial, with the result that at 12.947 indicated horsepower the consumption only averaged 1.59 pounds per indicated horsepower per hour. The contract stipulated that this trial should take place with a development of at least 12,200 horses, so that with the high power developed the low coal consumption is all the more creditable. But for the fact that the wind and sea were both too high, a speed trial at this power would have been made over a 12-knot course. From the records taken, however, it appeared certain that the speed at this power may be fairly taken to be about 17½ knots.

The full-speed trials took place March 23, on the selected deep-water course between Berry Head and Start Point, a distance of 12.25 knots. Four runs were made in alternate directions, the first and third being in the teeth of a northeasterly gale. Notwithstanding this, the mean speed realized was 18.3 knots. The speeds on the four runs were: First run, 17.92 knots; second

run, 18.08 knots; third run, 18.65 knots, and the last, 18.30 knots; the mean of the means being 18.30 knots, as before stated, with a mean indicated horsepower of slightly over 16,000. After the full-speed trial, circles were made to port and starboard with each steam-steering engine, the vessel meanwhile being at full speed. Maneuvering with the hand-steering wheels was also successfully carried out at a speed of 15 knots. The usual stopping, starting and reversing trials were made on the return voyage to Spithead, the return passage being made at a speed of 17 knots.—"Marine Review."

Iwate.—There was launched on the 29th of March, 1900, from the Elswick Shipyard of Sir W. G. Armstrong, Whitworth & Co., the Japanese first-class armored cruiser *Iwate*. The *Iwate* is a sister ship of the *Asama*, which was launched from the same yard some time ago. The principal dimensions of the vessel are: Length between perpendiculars, 400 feet; breadth, 68 feet 6 inches; depth, 41 feet; draught, 24 feet 3 inches; displacement, 9,750 tons. The armament consists of four 8-inch breech-loading guns twin mounted in barbettes; fourteen 6-inch quick-firing guns in 10-inch casemates, six on the main deck and four on the upper deck, the remaining four being on the upper deck protected by shields; twelve 12-pounder quick-firing guns, and eight 2½-pounder quick-firing guns. There are four submerged torpedo tubes, two forward and two aft. The vessel has a complete water-line belt of Harveyed nickel-steel armor, 7 inches thick amidships and reduced at the ends. Above this there is a citadel of 5-inch Harveyed nickel-steel armor inclosing the bases of the barbettes and carried from the top of the water-line belt to the main deck. The barbettes are of Harveyed nickel steel 6 inches thick. The casemates are of nickel-steel 6 inches thick. The conning tower is Harveyed nickel-steel 14 inches thick. The machinery is of the twin-screw, vertical, triple-expansion type, and is to develop 14,500 indicated horsepower. The speed guaranteed is 20¾ knots, and the boilers are of the Belleville latest type. The vessel has a bunker capacity for about 1,600 tons of coal. Accommodation is provided for an admiral, 52 officers and 430 petty officers and men.

NORWAY.

Norge.—There was launched on Saturday, March 31, from the Elswick Shipyard of Sir W. G. Armstrong, Whitworth & Co., the armor clad *Norge*, for the Norwegian Navy. This is the third warship built by the Elswick firm for the Norwegian Navy, while a fourth is at present on the stocks. The vessels previously launched were the *Tordenskjold* and the *Harald Haarfagre*. The *Norge* is 290 feet long, 50 feet 6 inches in breadth, 16 feet 6 inches draught, and her displacement in tons is 3,850, with a guaranteed speed of 16½ knots. She is to be armed with two 21-centimeter (8.26-inch) quick-firing guns, six 15-centimeter (5.9-inch), eight 12-pounders, six 3-pounders, and two torpedo tubes (submerged). The vessel has an armor belt of Harveyed steel 6 inches in thickness. The casemates, four in number, are of nickel-steel armor, 5 inches thick. The barbettes are of nickel-steel, 6 inches thick. The machinery is of the twin-screw, vertical, triple-expansion type, with Yarrow boilers, to develop 4,500 indicated horsepower.

RUSSIA.

Askold.—This protected cruiser was launched at the Germania works at Kiel on March 15, 1900. Her displacement is 6,500 tons, which will be increased at deep load to 7,900 tons. The protective deck is about 2¾ inches on the inclines, and 1½ inches scant on the flat. Two triple-expansion engines will give 19,000 I.H.P. and a speed of 23 knots, forced draft. The armament is entirely of rapid-fire guns, twelve 6-inch, twelve 3 inch and ten of 47-millimeter caliber; four torpedo tubes, two of which are submerged.

Cesarevitch.—The Russian battleship *Cesarevitch*, which is being built at La Seyne by the Forges et Chantiers de la Méditerranée, is of the same type as the French battleship *Jauréguiberry*. Her length is 361 feet; beam, 76 feet; displacement, 12,900 tons. She is built of Siemens-Martin steel, her upper armor deck being of the same material. The armor belt is 8 inches thick amidships, lessening to 4 inches fore and aft. Above the armor belt is a further protection, 6 inches in thickness,

reaching to the main deck. The protection of the casemates is 6 inches, and that of the barbettes 10 inches thick. She will carry four 12-inch, twelve 6-inch, and twenty 3-inch, and 28 smaller quick-firing guns, and also six torpedo tubes, two of them submerged. The *Borodino* and the *Orel*, building at St. Petersburg, are sister ships of the *Cesarevitch*, but are 8 feet longer, have 700 tons greater displacement, and the protection above the armor belt is an inch thicker. All three ships have 1½-inch protection against torpedoes. The engines develop 16,000 horsepower.

Som.—The destroyer *Som*, built and engined for the Imperial Russian Navy by Messrs. Laird Brothers, Birkenhead, has just completed a series of very satisfactory trials on the Clyde. The contract speed with the bunkers full and all outfit and stores on board was 27 knots, and the result on her official full-speed trial (February 24) gave a speed of nearly 28 knots as a mean of six runs on the measured mile, being nearly a knot in excess of the contract speed. The results on each of the six runs were as follows:

Miles.	Steam.	Time.		Speed.
		h.	*m.*	*Knots.*
1	225	2	11.2	27.44
2	223	2	10.4	27.61
3	223	2	7.0	28.35
4	219	2	8.8	27.95
5	215	2	9.8	27.73
6	210	2	12.0	27.27

The speed for the three hours' continuous steaming was 27.2 knots; the conditions under which this trial was run were unfavorable, owing to a heavy sea and strong southwest breeze. The Russian Government were represented by Captain Ouspensky, Naval Attache, Captain Dobrovolsky, commander of the vessel, Mr. Veshkourzev, naval architect, and Mr. Petroff, chief inspecting engineer. On February 28 and March 2 a series of progressive trials at varying speeds were carried out, as well as steering, stopping, starting and coal-consumption trials, all of which were successfully completed.—"Engineering."

MERCHANT STEAMERS.

Strombus.—In describing in our last issue the oil-fuel apparatus used on this large tank steamer we unfortunately appended a paragraph (last paragraph of the article, page 268,) wholly irrelevant to the description. As the burners used on the *Strombus* were described carefully in the previous paragraphs of the article, some confusion is created by this added description of an entirely different burner. Hence this first opportunity is taken to call our readers' attention to the fact that the burner described in the last paragraph of the article referred to was not used on the *Strombus* at all, but is a burner being experimented with, most satisfactorily, we understand, at the Armstrong-Whitworth works, with the very object of overcoming the noise and heat attendant upon the apparatus on the *Strombus*. The "note" was inadvertently included in the "copy" for *Strombus* article.

New 18,000-Ton Vessels for the Pacific Mail Steamship Co.—Work is progressing on the two magnificent steamers for the Pacific Mail fleet at the Newport News yard, and in a little more than a year from now these boats will be ready to contest the transpacific records. These two ships are of the largest size, and will be capable of making a sustained sea speed of 18 knots. Not only, indeed, will they be the largest and finest vessels on any route in the Pacific trade, but in point of size and equipment they will be in the same class with the best ships in the transatlantic service. The new vessels will be put on between San Francisco and Hong Kong, with Honolulu, Yokohama and Nagasaki as ports of call.

Following are the dimensions of the new vessels: Length between perpendiculars, 550 feet; length over all, 572 feet 4 inches; beam, 63 feet; depth, 40 feet; draught, 27 feet; displacement, about 18,600 tons. Accommodations will be provided for 200 first-cabin passengers, 30 white steerage and 1,200 Chinese.

Quarters for the latter are arranged so that the space may be utilized for other purposes, if unoccupied by Chinese.

The hulls will be constructed of steel, with frames spaced 32 inches apart throughout. A double bottom extends from stem to stern and is carried up to the turn in the bilge. There are four decks extending the whole length of the vessels, known as the lower, main, upper and promenade decks. In addition to these are the orlop and boat decks.

The main engines consist of two four-cylinder quadruple-expansion engines of the vertical, inverted, direct-acting type, placed abreast of each other in separate watertight compartments. The cylinders are of the following dimensions: H.P., 35 inches; 1st I.P., 50 inches; 2d I.P., 70 inches, and L.P., 100 inches diameter, with a common stroke of 66 inches. The engines are designed to develop 18,000 indicated horsepower, while running at 86 revolutions per minute. The cylinders are arranged in the following order: H.P. forward, L.P., 2d I.P., and 1st I.P. aft. The engine framing consists of cast-steel columns of I section, secured to the cylinders and bedplates. The bedplates are also of cast steel. The valves are of the single-ported piston type and are of cast iron. The link motion is the Stepheson link type. All the cylinders are steam jacketed. The piston rods are of forged steel fitted with cast-iron crosshead slippers lined with Parson's white bronze, which work in cast-iron crosshead guides. The connecting rods are of forged steel 138 inches long between center of crank and center of crosshead pin. The pistons are of cast steel and are of the dished pattern. The eccentrics are of cast iron keyed to the crank shafts. The eccentric rods and valve stems are of forged steel, the former being secured to the eccentrics by composition straps. Each main engine will be fitted with a steam and hydraulic direct-acting reversing engine and a 7-inch and 7-inch by 5-inch turning engine.

The thrust bearings will be two in number, of the ordinary horseshoe type, with fourteen shoes to each bearing. The stern-tube bearings will be of the ordinary brass sleeve pattern, fitted with lignum vitæ.

The crank shafts will be forged steel in four interchangeable

sections. The thrust, line and propeller shafts will also be forged steel. All shafts will be hollow. From the stern tube stuffing-box to the propeller hub the propeller shafts will have sleeves of composition, to protect against the corrosive action of the sea water. The propellers will be of the three-blade type, the hubs being cast steel and the blades bronze.

There will be two main condensers, one for each main engine. Each main condenser will have an independent air pump and two circulating pumps. The circulating pumps to be operated by a compound engine. Two auxiliary condensers will also be provided with combined air and circulating pumps attached. Two feed-water heaters will also be installed.

The boilers are to be nine in number, eight main boilers and one donkey boiler, of the horizontal, return fire-tube type. The arrangement of boilers is as follows: One donkey boiler on the upper deck in the boiler hatch. The main boilers placed in a fore-and-aft direction in two compartments, with all fire-rooms athwartship. In the after boiler compartment two single-ended and three double-ended boilers, and in the forward boiler compartment three double-ended boilers. Following are the principal boiler dimensions:

.....	Auxiliary boiler.	Main boilers.	
		Single-ended.	Double-ended.
Type	Scotch.	Scotch.	Scotch.
Number	1	2	6
Diameter, extreme, mean, feet and inches	11–6	16	16
Number of furnaces	2	4	8
Length of grates, feet and inches	6	5–9	5–9
Number of combustion chambers	1	4	8
Tubes, outside diameter, inches	2½	2½	2½
Heating surface, square feet	1,421	3,208	6,416
Grate surface, square feet	44	76.58	153.16
Ratio of heating surface to grate surface	32.3	41.89	41.89
Steam pressure, pounds	200	200	200

Ivernia.—The Cunard Company's new twin-screw liner *Ivernia*, built by Messrs. Swan and Hunter, Limited, and engined by the Wallsend Slipway and Engineering Company, Limited, both

with works on the Tyne, completed, at Liverpool, on the 27th ult., her deep-sea trials, which proved satisfactory alike as regards speed and seagoing qualities, for the weather in the North Sea was unpropitious. With the engines running at 90 revolutions, and indicating about 10,500 horsepower, the speed was 16.8 knots, whereas 16¼ knots was the contract stipulation. The ship is to be temporarily placed on the passenger service between Liverpool and New York, owing to some of the faster ships of the fleet being utilized as transports; but she is for the intermediate service of the company, and is the largest vessel yet constructed for this class of work. Her length over all is 600 feet, and between perpendiculars, 580 feet; beam, 64 feet 6 inches; depth molded to upper deck, 41 feet 6 inches; and gross tonnage, 13,900 tons. She has accommodation for 150 first, 200 second and 1,000 third-class passengers, and for 80 horses and 800 head of cattle, besides great holds—some of them insulated—for carrying frozen meat, etc. The engines, which we hope to illustrate later, are of the quadruple-expansion type, with cylinders 28½ inches, 41 inches, 58½ inches, and 84 inches in diameter by 54 inches stroke, supplied with steam at 210 pounds pressure from nine single-ended boilers worked on the Howden system.

New North German Lloyd Steamship.—The North German Lloyd line has just placed with the Vulcan Ship Building Co. of Stettin an order for a vessel which is to maintain an average speed of 25 knots an hour. In the contract it is further stipulated that the vessel shall have a length of 752 feet and that her engines are to develop 45,000 horsepower. Thus, in speed, horsepower and size this new leviathan will exceed all the others now afloat or building. The nearest approach to her in the way of speed is the Hamburg-American steamship *Deutschland*, which is soon to be placed in service in the transatlantic route, and which is guaranteed by her builders to have a sustained sea speed of not less than 23 knots an hour.

But the engines which are emplaced in the *Deutschland* have an indicated horsepower of only 33,000, as against 45,000 in the new vessel. In regard to size, this new product of the German

shipwright will far outclass anything now afloat. The *Oceanic* of the White Star line now holds the distinction of being the biggest ship that was ever launched, but her length over all is 704 feet, against the new vessel's 752 feet.

The Steamer Savoie, of the Compagnie Generale Transatlantique, was launched on March 13, at Saint-Nazzaire, France. The *Savoie* is said to be largest steamship ever constructed in a French shipyard. She is 533 feet long and 60 feet beam. Her tonnage is 10,500, and her engines are 12,000 horsepower, and are calculated to give her a trial speed of 22 knots. She will run on the line between Havre and New York.

YACHTS.

Gasoline Yacht Lady Francis.—In power-driven pleasure craft there is a noticeable increase in the number of vessels fitted with internal-combustion engines, which are of larger dimensions than the usual open launch. Nowadays the use of this form of motor is not infrequent in twin-screw vessels which in size and general appearance rank with small steam yachts. A vessel of this type recently completed for a Boston owner is the *Lady Francis*.

This vessel is 70 feet over all, 11 feet beam, and 4 feet 2 inches draught, with a gross tonnage of 30.16 tons, and net tonnage of 20.5 tons. She is fitted with twin screws, driven by independent gasoline engines, each developing 30-brake horsepower. In design the vessel is intended for service in the Atlantic coast. She has a full bow, easy running lines, with the greatest beam well aft, and a stern of the modified torpedo-boat type, thus increasing the water-line length and eliminating the tendency to yaw in a following sea. White oak is the material used for the frames, and cedar for the planking, which is double. The under diagonal course is $\frac{3}{8}$ inch thick, and the outer fore-and-aft course is 1 inch thick, with a layer of canvas prepared in beeswax and oil between. Copper fastening is used throughout, and all planking is riveted to the frames over copper burrs. There are two masts, rigged for fore-and-aft canvas.

Two four-cylinder self-starting gasoline engines are coupled to the screw shafts in the engine room aft. They are of the marine type, designed by the builders of the yacht. When starting, the initial impulse in the engine is given by a powder cartridge, which is slipped into a receiver on one of the cylinders. Exhausts from the engines are carried below the water line, so that objectionable noise and odor are got rid of. Each engine is equipped with an air pump, which supplies air to a tank under

a pressure of 55 pounds per square inch. The compressed air is used for a chime whistle and siren, operated from the pilot house. Separate engine telegraphs of the dial-and-pointer pattern are fitted in the engine room, the pilot house, and the upper steering platform. There are also speaking tubes, telephones and call bells throughout the boat for purposes of communication.

In the engine room a two-horsepower gasoline engine is direct-connected to a dynamo. The circuits are arranged so that the current can be supplied direct to the lamps and searchlight or to charge a storage battery, the batteries being placed under the floor. Small electric lamps are fitted to light the telegraph dials and the binnacles, and deck plugs and extension lamps are provided for deck use. The gasoline tank has a capacity for sufficient fuel to last on a thousand-mile run.

Steel is used for the propeller shafts, and these are fitted with bronze sleeves. The wheels are of bronze also, and are four-bladed, 34 inches diameter.

Two boats are carried—one 14-foot dinghey on the starboard side, and one 14-foot motor boat with 1 horsepower engine on the port side.

Both the hull and machinery were designed by Charles B. King, of Detroit, Michigan, and the vessel built by the Charles B. King Co., of that city.—"Marine Engineering."

ASSOCIATION NOTES.

The following letter from the Institution of Civil Engineers, England, has been received by the Secretary, and has been duly acknowledged. The invitation cards referred to as being enclosed were directed to the President and the Secretary of the Society.

A list of the members and associates of this Society has been forwarded to the Secretary of the Institution, so that any who may find it possible to be present at the annual can readily secure individual cards of invitation after arrival in London by communicating with the Secretary of the Institution.

"THE INSTITUTION OF CIVIL ENGINEERS,
GREAT GEORGE STREET, WESTMINSTER, S. W.

May 9, 1900.

"A. B. WILLITS, ESQ., *Secretary*,
"AMERICAN SOCIETY OF NAVAL ENGINEERS,
"Navy Department, Washington, D. C., U. S.

"My Dear Sir: In respect of the visit of members of your Society in July, which is anticipated here with so much pleasure, I am desired by the President to inform you, and to request you to make it known to the members of the American Society of Naval Engineers, that he proposes to formally welcome the visitors in the Institution Library, at 3 o'clock, on the afternoon of Monday, the 2d July. Thereafter, during that week, the accommodation of the Institution rooms will be entirely at the disposal of your duly-accredited members; and certain fixtures of a social kind are now under consideration by our Reception Committee, to attend which invitations will be sent to the visitors, at their addresses on this side, which they should intimate to me as soon after their arrival as convenient to them, if they can not do so previously.

"With regard to details of the arrangements referred to, I hope to be in a position to inform you further presently.

"Meanwhile, I am desired by the President to send you, by registered packet, two invitation cards to the Institution Annual Reception, on the 5th July, made out to the office bearers of your Society, and to request you to be so obliging as to add their addresses on the envelopes and to distribute them. The names I should point out are taken from the most recent lists we have. Kindly let me know if any are overlooked.

"In confiding these invitations to your kind offices at this stage, I am to point out that it is desired thus to convey some special mark of courtesy to the Council of the Society; and it will, of course, be understood that those of your members who do the Institution the honor of attending at the time of the Reception will be equally cordially invited when the President is made aware of their names.

"Believe me to be, yours faithfully,

"J. H. T. TUDSBERY, *Secretary*."

ERRATUM.

Page 311, Vol. XII (May, 1900), contract trial *Albany's* piston speed, feet per minute, should read 807, instead of 747, as given.